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Dynamic Measurements of a Novel System

Combining Natural Ventilation with Diffuse Ceiling Inlet and Thermally Activated Building Constructions

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DEPARTMENT OF CIVIL ENGINEERING
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DCE Technical Report No. 194

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by

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1. Introduction

In the PSO project 345-061, a novel system solution combining natural ventilation with diffuse ceiling inlet and thermally activated building systems (TABS) has been proposed for cooling and ventilation in Danish office buildings. Due to the application of diffuse ceiling inlet, cold outdoor air can be supplied into the room without any risk of draught even in the extreme winter. This means that natural ventilation is available even in winter and it is beneficial to reduce the energy consumption for buildings with cooling demand in cold seasons. The highly energy-efficient TABS can be activated to supplement the extra heating/cooling when the thermal comfort cannot be satisfied. In this way, the building systems can operate at a very low energy use all the year round. The schematic diagram of this new solution is shown in Figure 1, details of the operation principle can be found in Ref. [1].

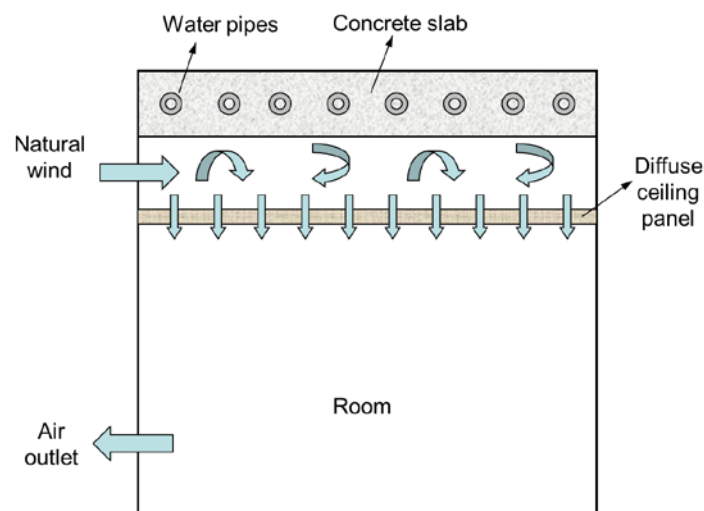


Figure 1 Schematic diagram of the new system solution [1].

Since the ceiling panel behaves as an insulation layer, this ceiling panel may have an influence on the heat transfer between TABS and room space. Moreover, the heat transfer in the plenum is very complicated due to the combined effect of ventilation and TABS as well as diffuse ceiling. Therefore, to test this combined system and study the issues mentioned above, an experiment set-up was built up in Aalborg University.

Initially, the steady-state tests were carried out, in order to study the energy performance of TABS with and without the effect of diffuse ceiling. Lower supply water temperatures were used for TABS with ceiling panel under both cooling and heating modes, compared with those of cases without ceiling panel. Therefore, ceiling panel is beneficial to TABS heating, but decreases TABS cooling. Water temperatures used are too low for TABS cooling with diffuse ceiling, below 15 °C in most cases, which is impractical. Thus, dynamic thermal process may be different, and higher water temperature can be used for energy storage before the occupied time.

The previous test on the response time of TABS shows that the response time of TABS used in our case is almost 5 hours, which only reaches 63% of the total cooling capacity. It needs more time to get the full cooling capacity, so the steady-state condition cannot represent the real thermal processes in the practical applications.

Further, the building thermal performance is dynamic depending on the weather conditions and occupations. TABS may be activated outside the working time (during the night or other unoccupied time) for energy storage. Moreover, different control strategies should be used depending on the weather conditions and occupations.

Therefore, based on the necessities mentioned above, the dynamic measurements are designed. The main objectives of the dynamic measurements are to investigate the dynamic behavior of the system and to test different selected control strategies. The measurements profiles for outdoor temperature, solar radiation and internal heat load are defined for three typical days (winter, spring/autumn and summer) in Denmark. The main control parameters will be ventilation rate and water temperature/flow rate of TABS. The control strategy should include both passive and active use of TABS, and get the utmost utilization of cooling potential from natural ventilation.

In this report, the dynamic measurements are presented, and results of energy performance of the system and thermal comfort in the test room under different control strategies are discussed. The tested results will give some suggestions related to the control strategies under different conditions, which is beneficial to the future applications of this combined system in office buildings.

2. Experimental set-up

The same experimental set-up in the Hotbox for steady-state tests is used in the dynamic measurements, details about the Hotbox can be found in the steady-state experiment report [2]. But a new dynamic control system is developed for the flexible parameter control, including the control of internal heat sources, solar radiation, ventilation, window opening, TABS, and so on. The internal heat sources have an On/Off control using a timer. Solar radiation is simulated by an electric carpet, and controlled by a dynamic voltage input. Ventilation rate is controlled by a mechanical fan and a frequency transformer, and a voltage output based on the room air set-point is sent to the frequency transformer. Window opening has an On/Off control using a timer, so does TABS. The updated set-up in the room is shown in Figure 2.



Figure 2 Set-up in the room.

3. Designed dynamic experiments

Based on the Danish Reference Year [3], three typical climatic conditions- winter, transitional seasons and summer are considered in the dynamic measurements. Table 1 shows the detailed conditions in the typical weather. The cold box can simulate the air temperatures of these typical

conditions periodically, and the solar radiation is simulated by an electric carpet located on the floor as in the steady-state measurements. Both the cold box air temperature and the solar radiation are considered to reach the maximum at 13:00. Surrounding and upper zones have the same constant air temperature during all day. The occupied time considered is from 9:00 to 17:00, so the internal heat sources with a level of about 30 W/m² are switched on during this period.

Two occupants are considered in the room, and the minimum ventilation for the requirement of indoor air quality is 72 m³/h (36 m³/h per person). This minimum ventilation rate corresponds to an air change per hour of about 1.7 h⁻¹ (ACH=1.7). The maximum ventilation rate is flexible in each case in order to use the ventilation cooling at the maximum level, but this maximum should not be higher than 387.8 m³/h (ACH=9) since there is a limitation of the fan control. The ventilation is carried out by the air circulation between the cold and hot chambers with the help of a mechanical fan, and the principle can be found in Ref. [2].

Table 1 Designed conditions for dynamic measurements.

Case number	Typical days	Cold box air temperature (°C)	Internal heat load during occupied hours (9-17) (W/m ²)	Solar radiation	Surrounding/upper zone air T (°C)
Case 1	Winter (average January)	-1.0 (±2.5)	30	0 (cloudy sky)	22
Case 2	Transitional seasons (average May)	10.9 (±5.75)	30	Changing (3432 Wh/m ² window area/day)	23
Case 3	Summer (average July)	20.0 (±6)	30	Maximum (clear sky) (4848 Wh/m ² window area/day)	24

In order to ensure a good control, the air temperatures in the cold box are designated with a sinusoidal pattern described by the Equation (1). Figure 3 shows the designed air temperatures in the cold box.

$$T = A + B \times \sin\left(\frac{\pi \times t}{12 \times 60} - \frac{5 \times \pi}{12}\right) \quad (1)$$

Where, A and B are terms for the three typical conditions, t is the time in minute,

Winter: A=-1, B=2.5

Transitional seasons: A=10.9, B=7.75

Summer: A=20, B=6

The solar radiation in Table 1 is the radiation projected to the window, and the calculation of solar heat gain into the room is shown below:

For transitional seasons,

$$\frac{3432 \times 1.4 \times 2.4 \times 0.85 \times 0.5 \times 0.8}{15.84} = 247.52 \text{ (Wh/(m}^2 \text{ floor} \cdot \text{day))}$$

For summer,

$$\frac{4848 \times 1.4 \times 2.4 \times 0.85 \times 0.5 \times 0.8}{15.84} = 349.64 \text{ (Wh/(m}^2 \text{ floor} \cdot \text{day))}$$

Where, the net glass area ratio to window is 0.85, the g-value is 0.5, and shading factor is 0.8. The dynamic curves of solar radiation are shown in Figure 4.

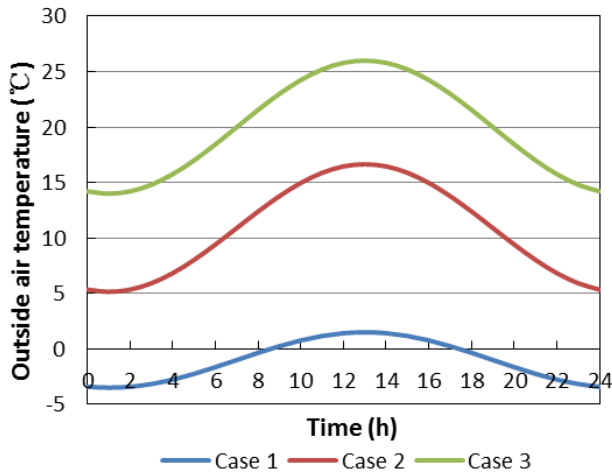


Figure 3 Outside air temperature.

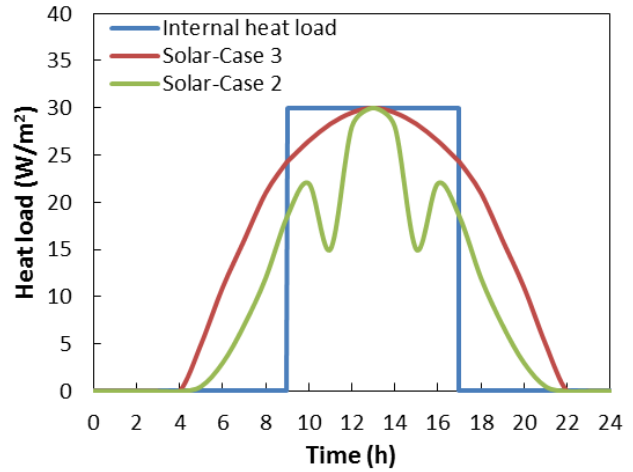


Figure 4 Heat load level in the room.

When TABS are activated in the dynamic measurements, the water flow rate is kept at a constant of 0.037 kg/s, which corresponds to the recommended water velocity for this system in our case. For the heating case, the supply water temperature is kept at 32 °C, while it is kept at 16 °C for the cooling condition. Therefore, only the operating time is changed in the measurements.

4. Measurements

The dynamic measurements are performed under the quasi-steady state. The results can be acceptable only when each parameter gets almost the same value at the same time of two days.

4.1 Measured parameters

The primary measuring parameters include air temperatures, surface temperatures, water flow rate and temperatures, ventilation air flow rate, air velocity, and power of heat sources. The detailed measuring points can be found in the last steady-state measurements [2]. Everything keeps the same as in the last measurements, except the measuring positions are changed, and 6 measuring columns are placed at the fixed positions as depicted in Figure 5.

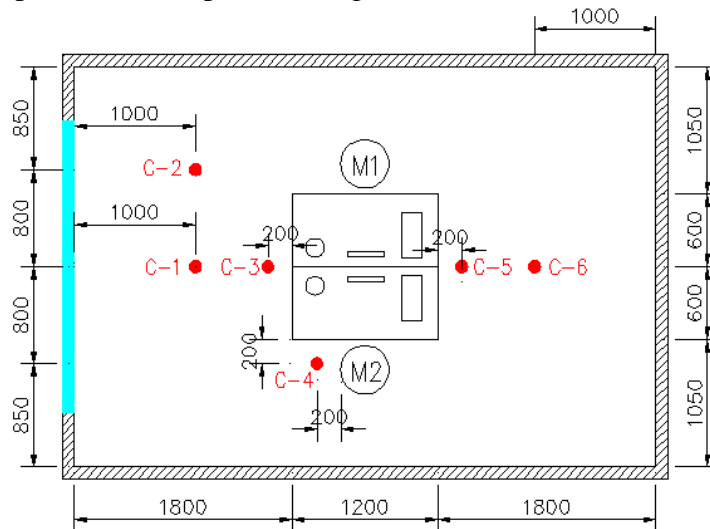


Figure 5 Measuring positions in dynamic measurements.

The anemometers are calibrated before the tests, so they can be used to measure both the air temperature and the air velocity at the same position. Each column has 5 anemometers at the height of 0.1 m, 0.6 m, 1.1 m, 1.7 m and 2.3 m.

4.2 Measurement devices

Table 2 lists all details of the devices used in the measurements.

Table 2 Details of measurement devices used.

Devices	Type	Measurements	Number	Remarks
Data-logger	Fluke Helios Plus 2287A	Temperature, Pressure drop, humidity	1	Together with 2 compensation boxes
Micro-manometer	FCO510	Pressure drop	1	Between locations in the plenum and room
Brunata		Water mass flow rate	1	
Precision Multimeter	PREMA 5017 Precision Multimeter	Voltage/Power of carpet	1	Together with the calibrated carpet to measure the power
Danteck Anemometer and Datalogger	DANTECK DYNAMICS	Air velocity and air temperature	2	
Power supply	Labor-netzgerat Power supply EA-3013S	Power supply for measuring humidity	1	
Voltage transformer			3	
Precision power meter	Fluke 345 PQ clamp meter	Power of internal heat sources	1	
Differential pressure transducer	FCO 44	Pressure difference of orifice	1	
Mechanical fan			1	
Micro-monometer		Ventilation rate	1	
Heat sources	2 manikins, 2 desk lights, 2 computers, 2 monitors			

5. Control strategies in different climatic conditions

As described at the beginning, the main control parameters will be air flow rate and water temperature/flow rate of TABS. The control strategy should include both passive and active use of TABS, and get the utmost utilization of cooling potential from natural ventilation. Since in different seasons the weather conditions are different and influence the cooling effect of ventilation and thermal behaviour of TABS. Therefore, the control strategies will change with the weather conditions. The basic control will be based on both room air temperature and outside air temperature.

When there is a heating need of TABS during the occupied hours, ventilation rate will be kept at the minimum, and TABS are activated for heating. When there is a cooling need of TABS during the occupied hours, if the outside air temperature is suitable for cooling then the ventilation rate can be changed to the maximum, meanwhile, TABS are activated for cooling. However, if the outside air temperature is higher than room temperature, then the ventilation rate should be kept at the minimum and TABS are used for cooling.

The initial control strategies used in the dynamic measurements are designed in Figures 6-8.

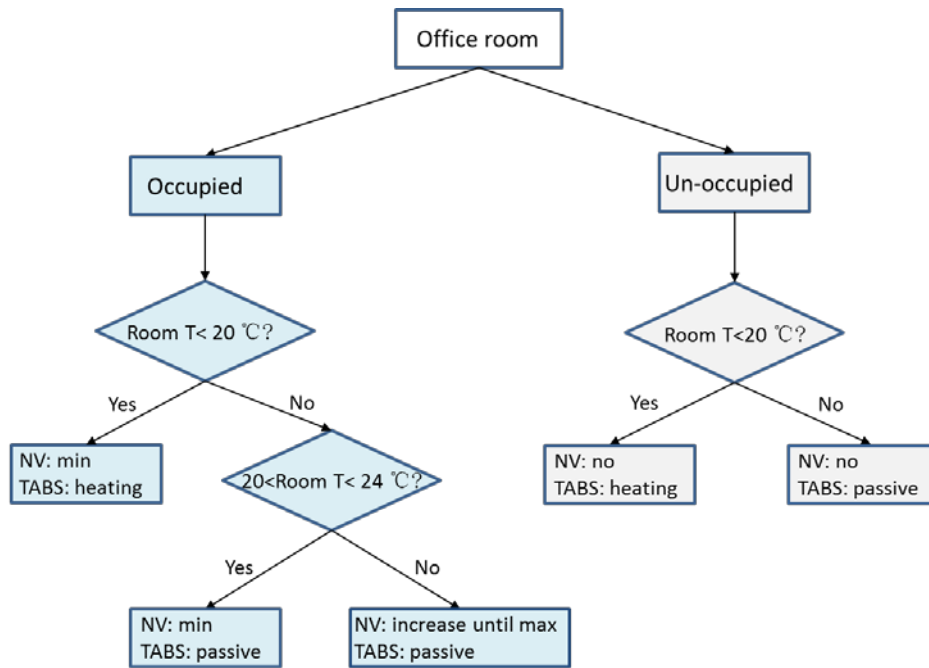


Figure 6 Flow chart of control strategy in winter.

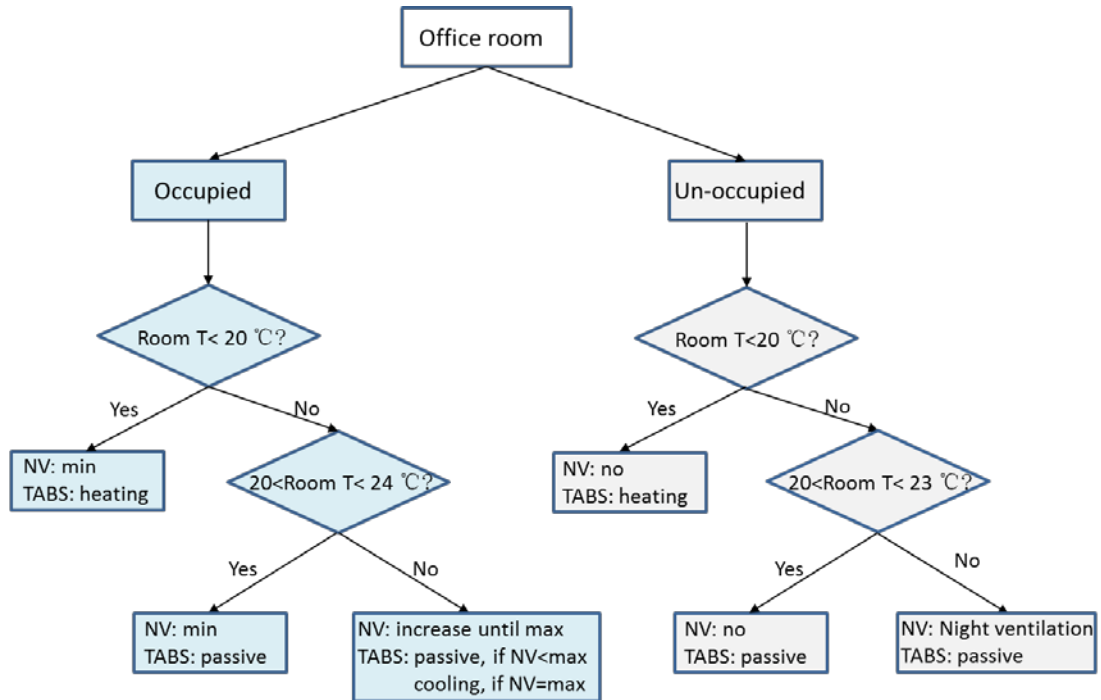


Figure 7 Flow chart of control strategy in spring/autumn.

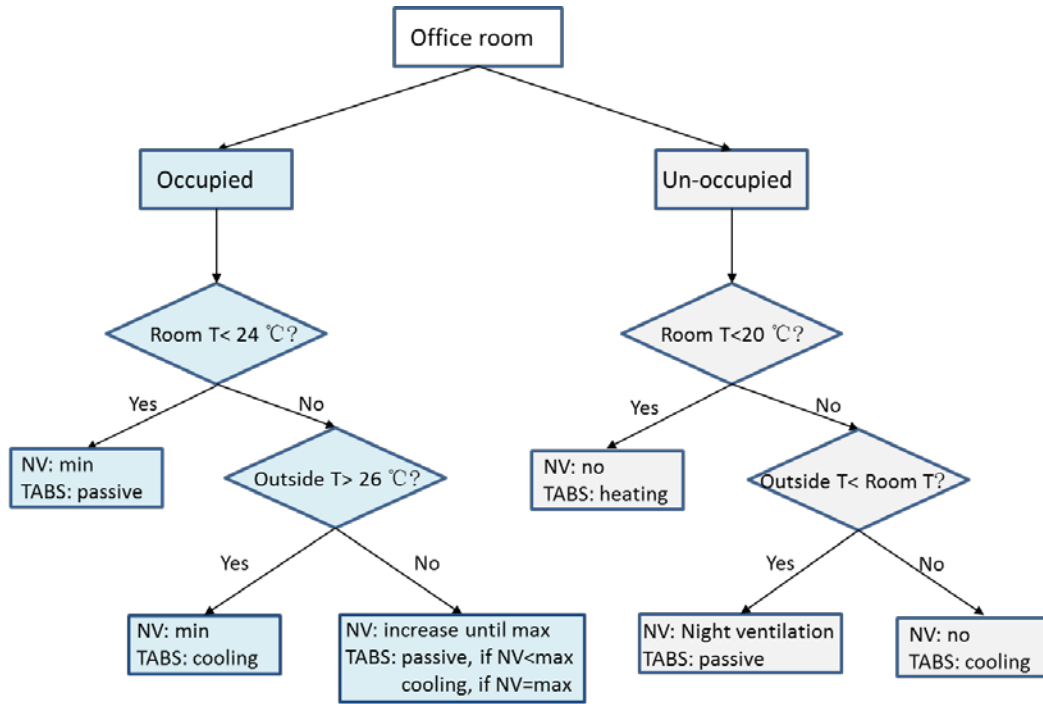


Figure 8 Flow chart of control strategy in summer.

The above flow charts show control strategies in different seasons, and the main objective in the control strategies is to use natural ventilation and passive TABS to the maximum level. Finally, the least energy can be consumed. When TABS are to be used from the morning, it is better to activate the system some hours before the occupied hours, since TABS have relatively longer response time.

6. Experimental results

The data is recorded every second, and the results are analysed every 10 seconds.

6.1 Case 1 – Typical winter

Case 1 started from 6/9/2015 to 6/27/2015, using the basic control strategy in Figure 6, but two statuses are considered. Firstly, no TABS heating is used even the room temperature is lower than 20 °C, the measuring time is from 6/9/2015 to 6/18/2015. Secondly, there is TABS heating between 6:00 and 8:00 for two hours, and the measuring time is from 6/18/2015 to 6/27/2015. The purpose of testing two statuses is to compare the thermal environment and the energy performance under these two conditions.

6.1.1 No TABS heating (6/9/2015-6/18/2015)

The process of getting quasi-steady state in this case is given in Figures 9 and 10, and the results get stable after 8 days.

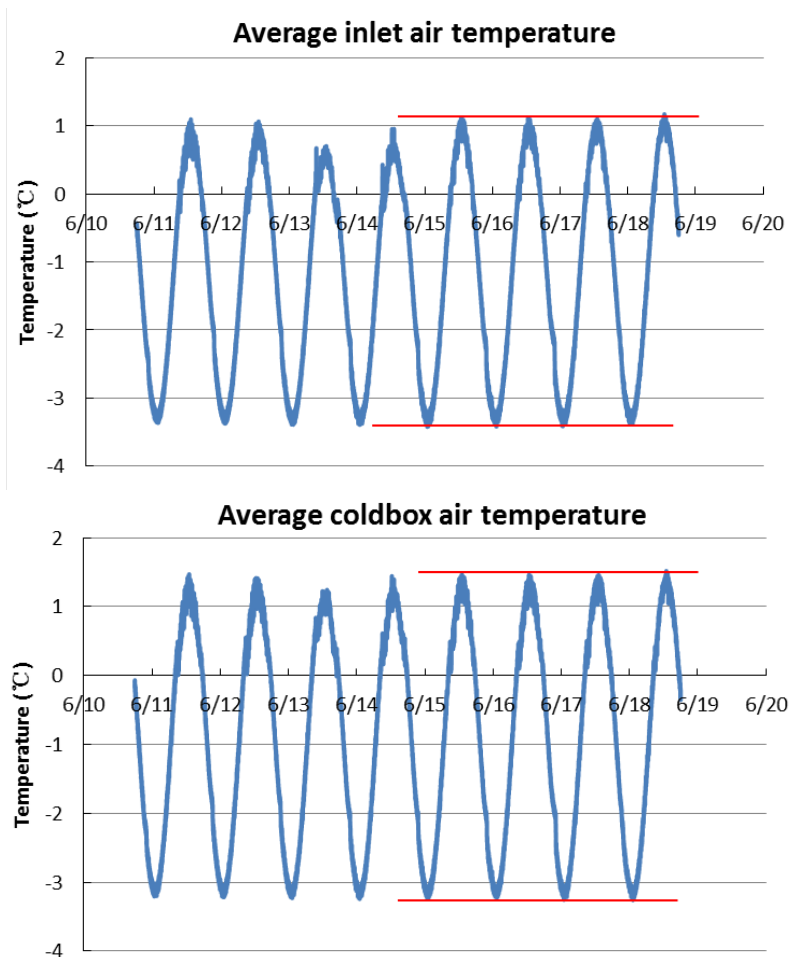
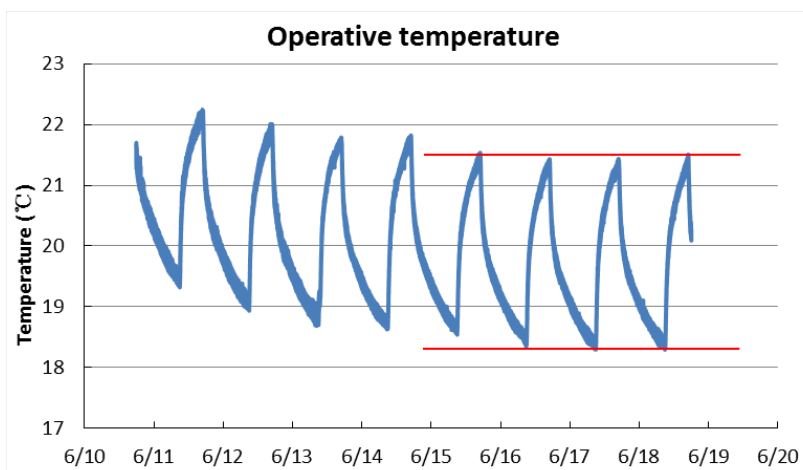


Figure 9 Cold box conditions.

The air temperatures of the three inlets have a maximum different of about 0.3 °C during the day and 0.5 °C during the night. Since the air distribution in the cold box is not so even, the inlet air temperature is about 0.3 °C lower than the cold box air temperature.



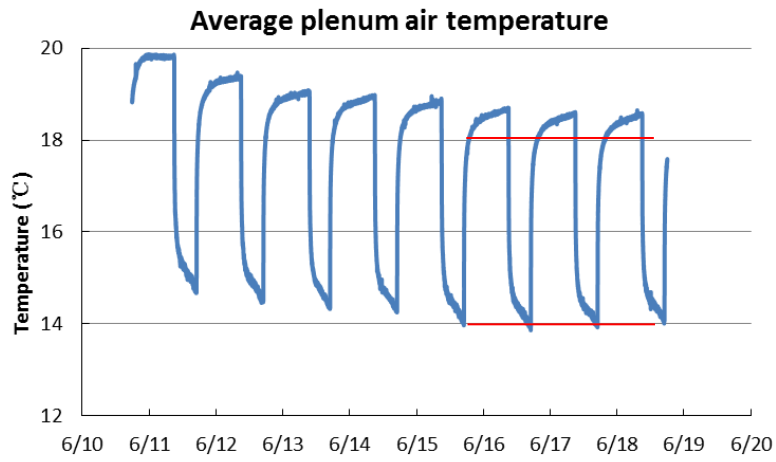
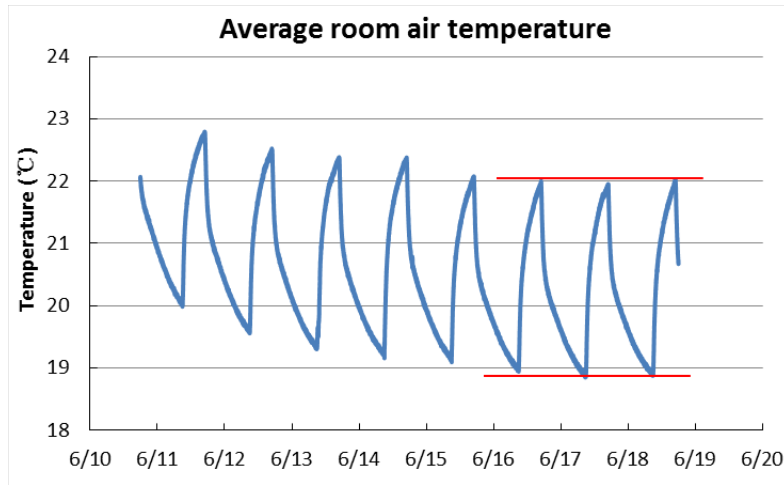


Figure 10 Room side conditions.

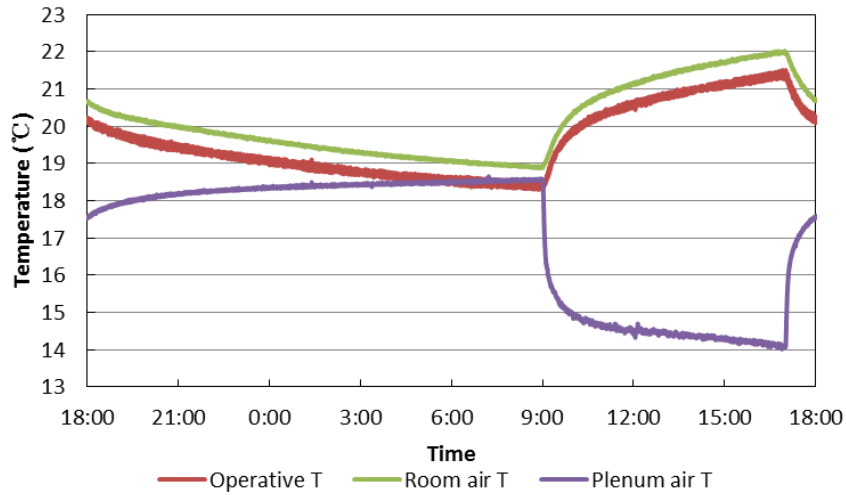


Figure 11 Quasi-steady state results of Case 1-no TABS heating.

Figure 11 shows the results of the last day, when the system gets the quasi-steady state. The operative temperature measured by the two global sensors is between 18.4-21.4 °C during the occupied hours, the minimum occurs at 9:00 in the morning. This minimum temperature is out of the comfort range in the winter [4]. The room temperature during the un-occupied hours is also lower than 20 °C. Therefore, it is better to activate TABS in the morning to provide the extra heating to the room, then the temperatures can be increased.

The plenum air is between 14.0-18.5 °C during the whole day, and the minimum occurs at 17:00. The low air temperature in the plenum makes it very efficient to use TABS heating. According to results in the steady-state measurements, this low plenum air temperature is beneficial to TABS heating.

Table 3 shows the control strategy used in Case 1 without TABS heating, the ventilation rate is always kept at the minimum level with the measured value of 70.9 m³/h.

Table 3 Control strategy used in Case 1 without TABS heating.

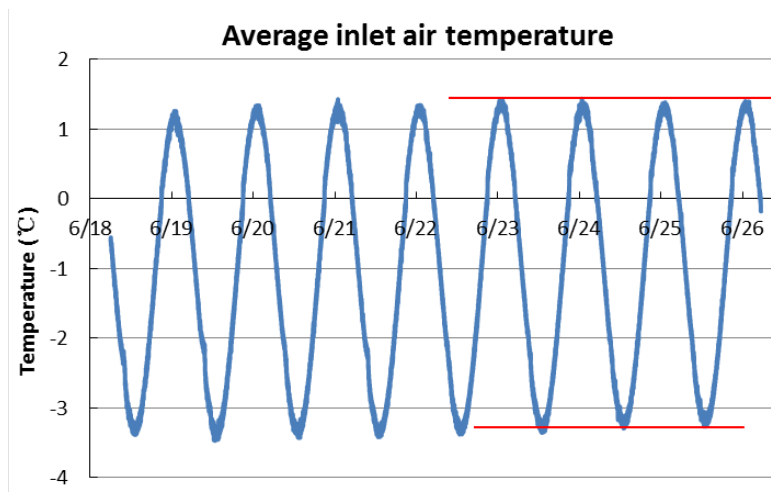
Time	Ventilation	TABS heating	Internal heat sources	Solar radiation
9:00	On (Min)	Off	On	Off
17:00	Off	Off	Off	Off

Overall, for Case 1 if there is no TABS heating, the temperatures in the lower room have the minimum at 9:00. Therefore, it is better to activate TABS for several hours before the occupied hours.

6.1.2 With TABS heating (6/18/2015-6/27/2015)

From the results above, it can be seen that the room temperature is out of the thermal comfort range if there is no extra TABS heating in the winter case, and it is the best to activate TABS before the occupied hours. Therefore, from the morning of 6/18/2015, TABS are activated for two hours from 6:00 to 8:00. The measured supply water temperature is 31.6 °C, and the constant water flow rate is 0.036 kg/s.

The process of getting quasi-steady state in this case is given in Figures 12 and 13, and the results get stable after about 6 days.



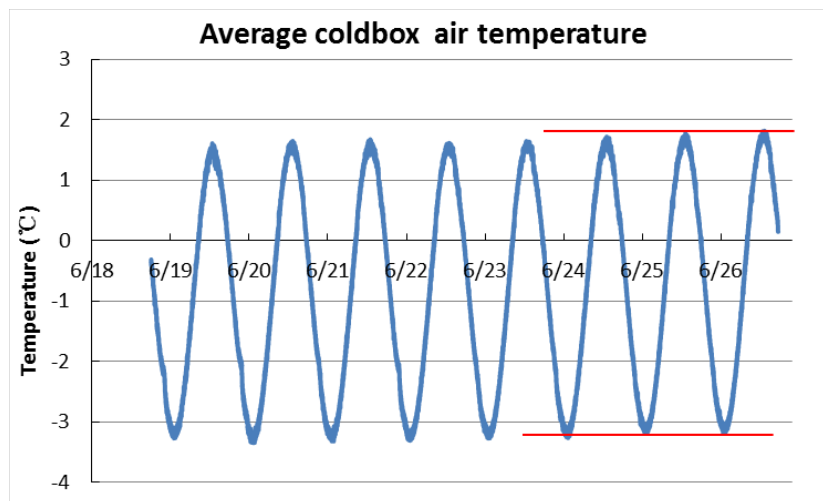
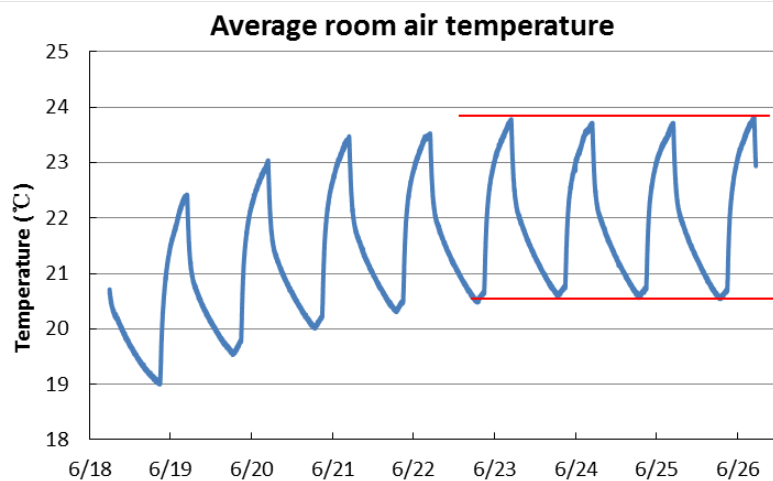
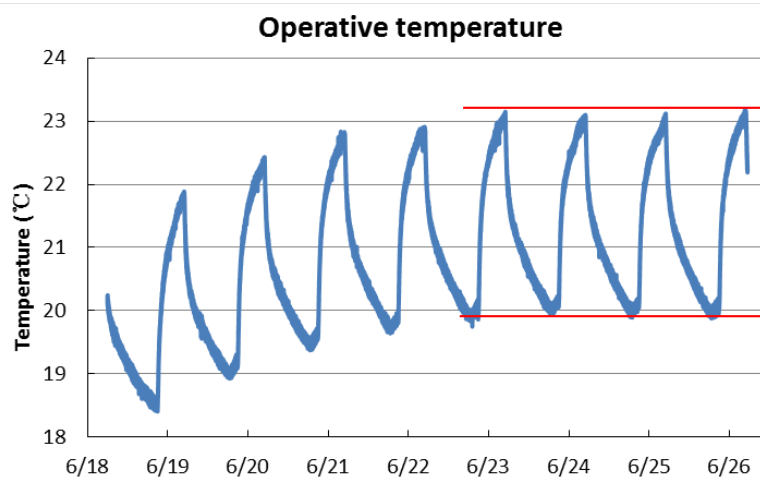


Figure 12 Cold box conditions.



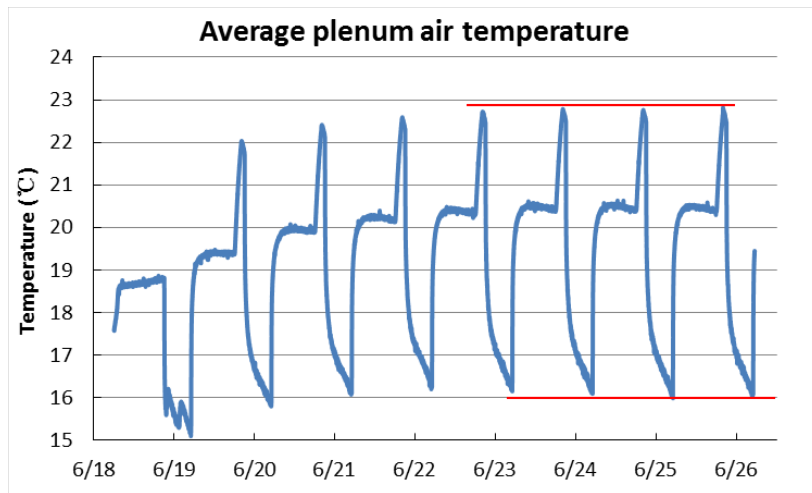


Figure 13 Room side conditions.

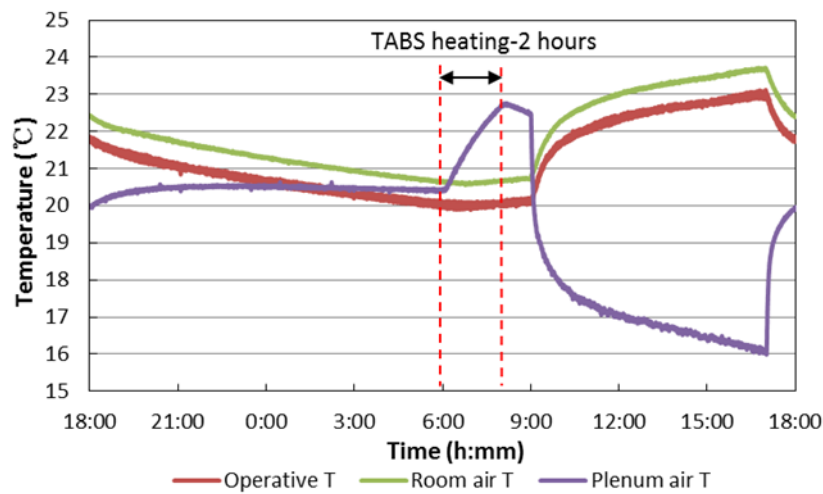


Figure 14 Quasi-steady state results of Case 1-with TABS heating.

Figure 14 shows the results of last day, when the system gets the quasi-steady state. The operative temperature measured by the two global sensors is between 20.0-23.0 °C during the occupied hours, the minimum occurs at 9:00 in the morning. This minimum temperature is within the comfort range in the winter [4]. The room temperature during the un-occupied hours is higher than 20 °C. Compared with the results in Figure 11, it can be seen that the results are improved when TABS are activated from 6:00 to 8:00.

Figure 15 shows the supply and return water temperatures of TABS. The activation of TABS takes the system approximate 6 days more to get the quasi-steady state.

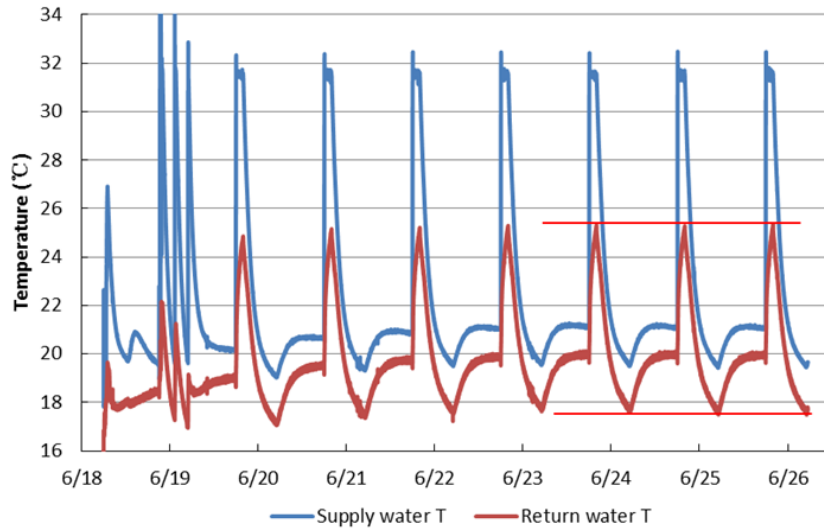


Figure 15 TABS water temperatures.

Table 4 shows the control strategy used in Case 1 with TABS heating, the only difference compared with Table 3 is the activation of TABS from 6:00 to 8:00.

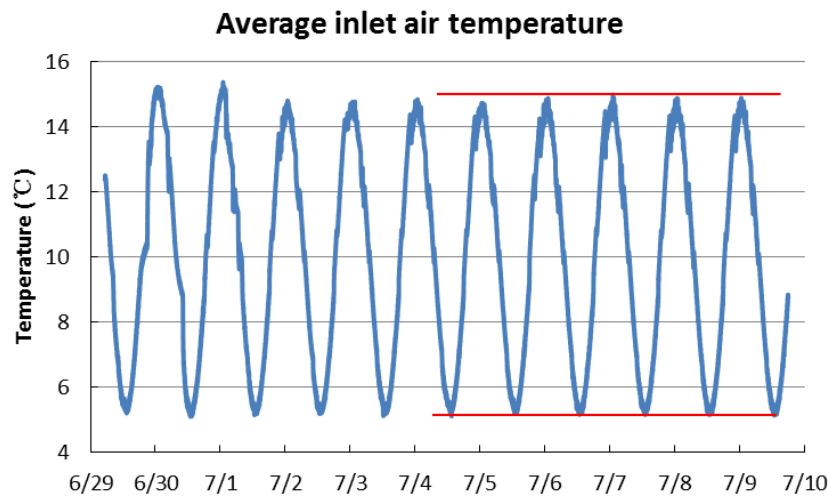
Table 4 Control strategy used in Case 1 with TABS heating.

Time	Ventilation	TABS heating	Internal heat sources	Solar radiation
6:00	Off	On	Off	Off
8:00	Off	Off	Off	Off
9:00	On (Min)	Off	On	Off
17:00	Off	Off	Off	Off

6.2 Case 2 – Typical transitional seasons

Case 2 started from 6/28/2015 to 7/9/2015, dynamic operation of the system without any mechanical cooling under typical spring\autumn conditions is measured. The main idea in Case 2 is to use natural ventilation to the maximum, and the basic control strategy in this case is depicted in Figure 7.

The process of getting quasi-steady state in this case is given in Figures 16 and 17, and the results get stable after about 10 days.



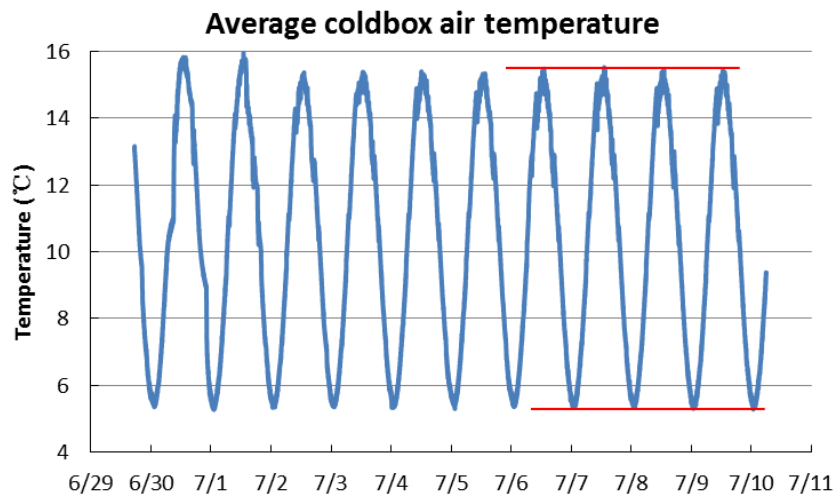
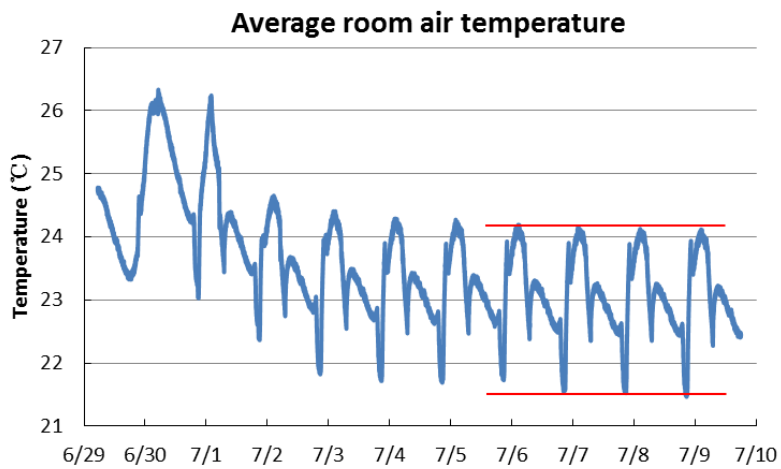
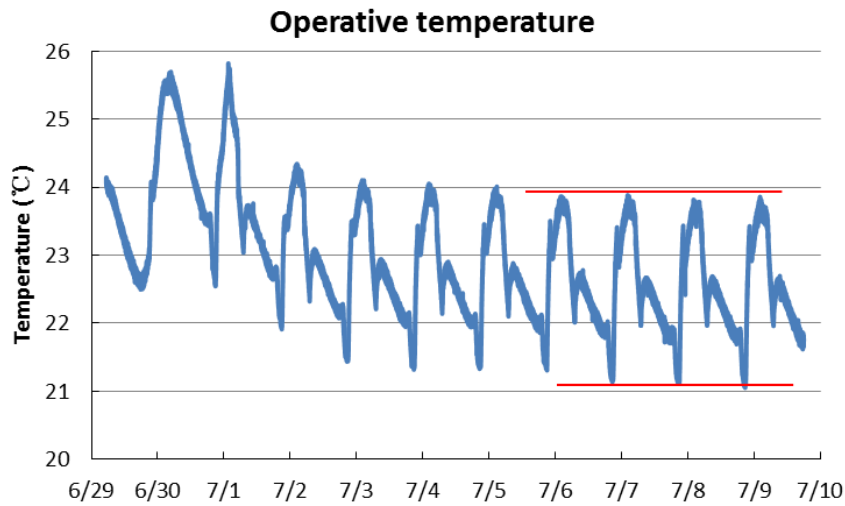


Figure 16 Cold box conditions.

In Case 2, the temperatures of the three inlets only have a maximum difference of about 0.2 °C, so the air in the cold box is more evenly distributed.



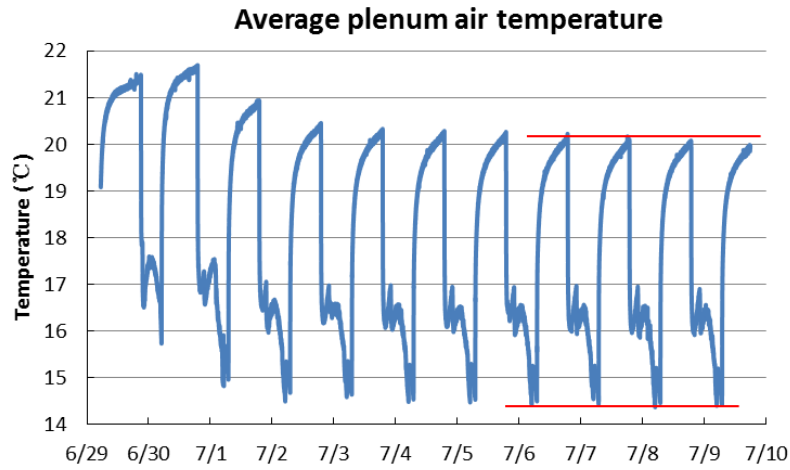


Figure 17 Room side conditions.

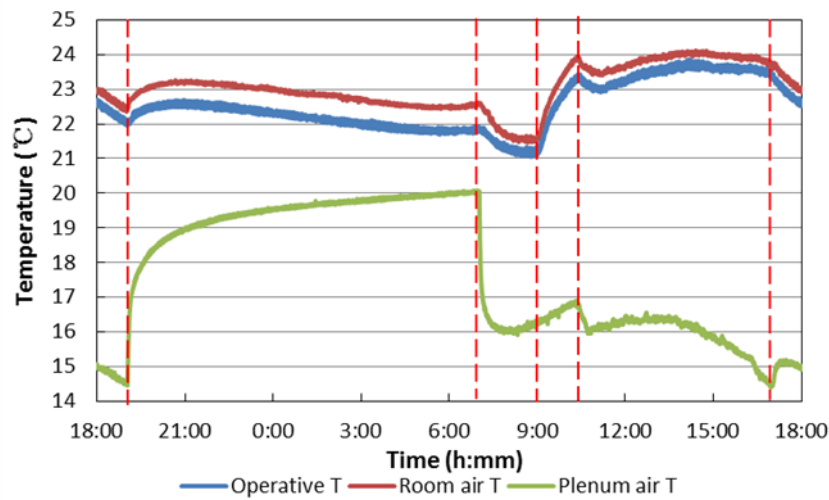


Figure 18 Quasi-steady state results of Case 2.

Figure 18 shows the results of the last day, after the system gets the quasi-steady state. Table 5 gives the control strategy used in Case 2. It can be seen in Figure 18 that this control strategy can keep the operative temperature within the range of 21.0–24.0 °C. The ventilation time is from 7:00 to 19:00, since during this period the solar heat gain increases the room temperature. The climatic condition in the transitional seasons is very suitable for natural ventilation, during the occupied hours when the room temperature is above 24 °C, it is very efficient to increase the ventilation rate to use the natural cooling. The measured ventilation rates are 145 m³/h and 285.4 m³/h for the low and high levels, corresponding to approximate 2 times and 4 times the minimum ventilation rate. In order to have a better control, the mechanical fan is controlled using a step setting rather than a continuous regulation. This means the ventilation rate changes from 145 m³/h to 285.4 m³/h directly, and keeps the constant ventilation rate during the corresponding period. The ventilation rate is initially determined by a simple heat balance estimation.

Table 5 Control strategy used in Case 2.

Time	Ventilation	TABS	Internal heat sources	Solar radiation
7:00	On (2×Min)	Off	Off	On
9:00	On (2×Min)	Off	On	On
10:15	On (4×Min)	Off	On	On
17:00	On (2×Min)	Off	Off	On
19:00	Off	Off	Off	On

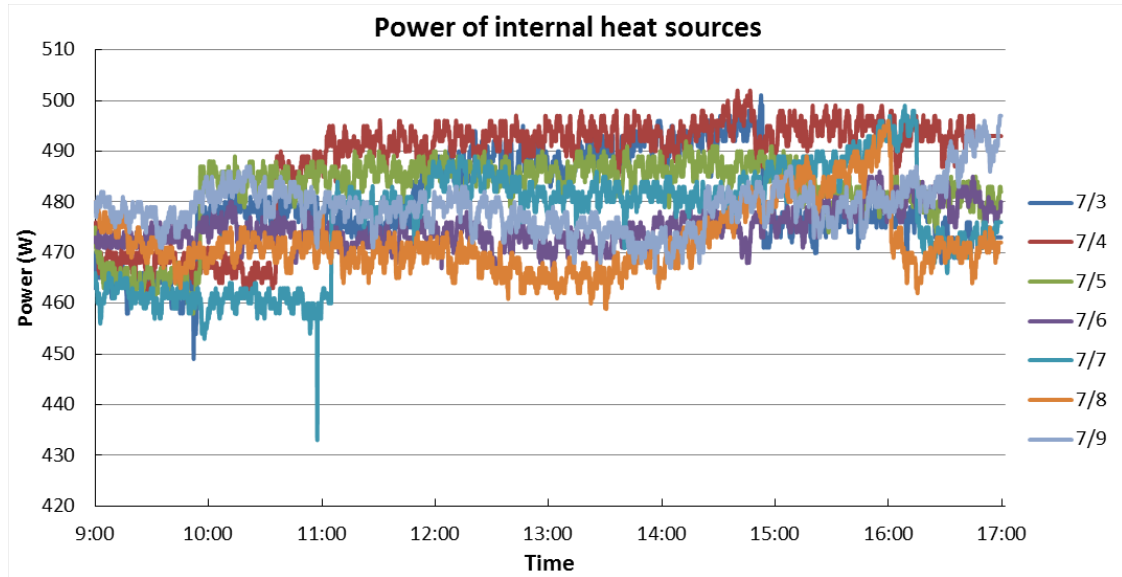


Figure 19 Power consumption of the internal heat sources.

Figure 19 shows the power consumption of internal heat sources. Since the voltage in the lab slightly varies during the day and night, the power consumption of the equipment changes smoothly. As shown in Figure 19, at the early occupied hours the power consumption is relatively lower, and it increases with time. The average power consumption of all internal heat sources is about 480 W, corresponding to approximate 30.3 W/m^2 floor area.

Figure 20 depicts the power consumption of the electric carpet representing the solar radiation. Due to the changing voltage in the lab, this power consumption is not stable with smooth changes.

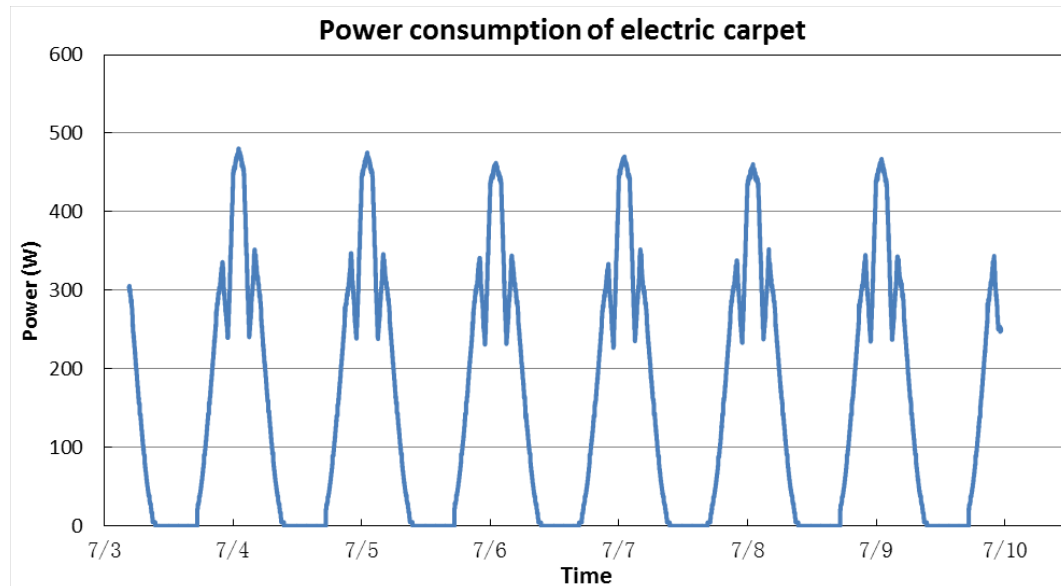


Figure 20 Power consumption of electric carpet.

6.3 Case 3 – Typical summer

Case 3 started from 7/11/2015 to 7/22/2015, the basic control strategy is given in Figure 8. Due to the long response time of TABS, it is better to activate TABS cooling before the occupied hours. In this case, it is activated from 7:00 in the morning.

The process of getting quasi-steady state in this case is given in Figures 21 and 22, and the results get stable after about 6 days. Since the ventilation rate is relatively large, the system gets the quasi-steady state very fast.

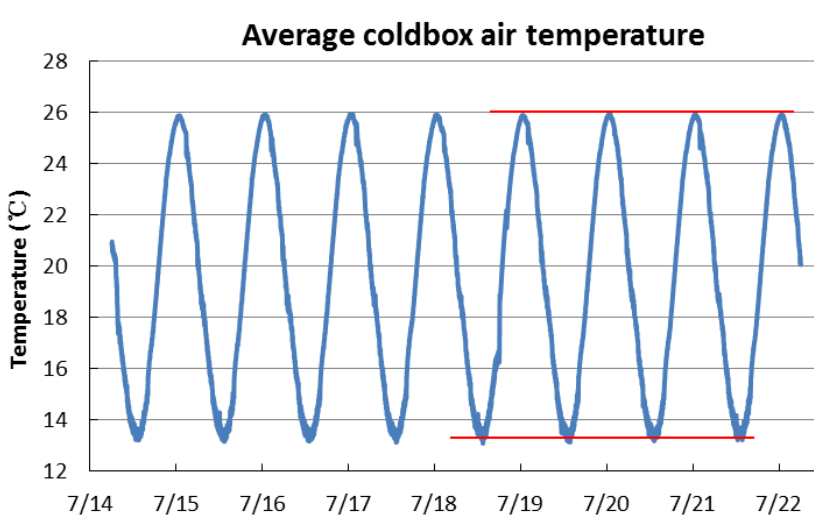
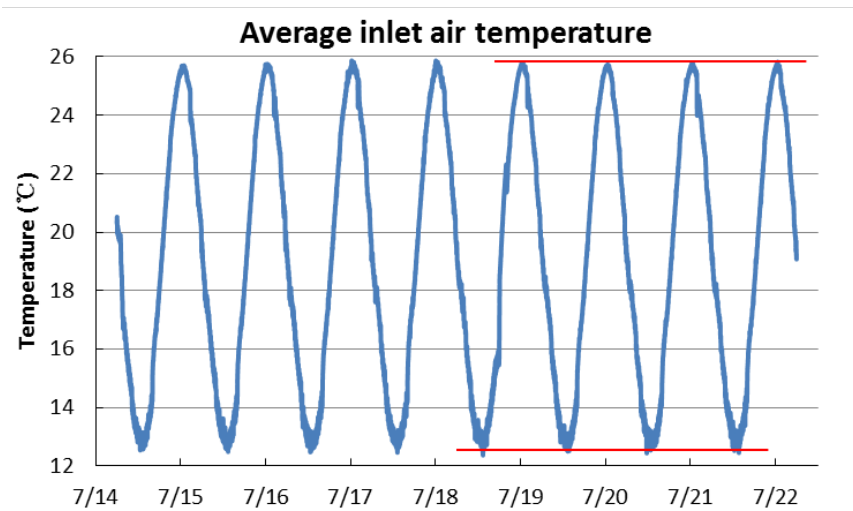
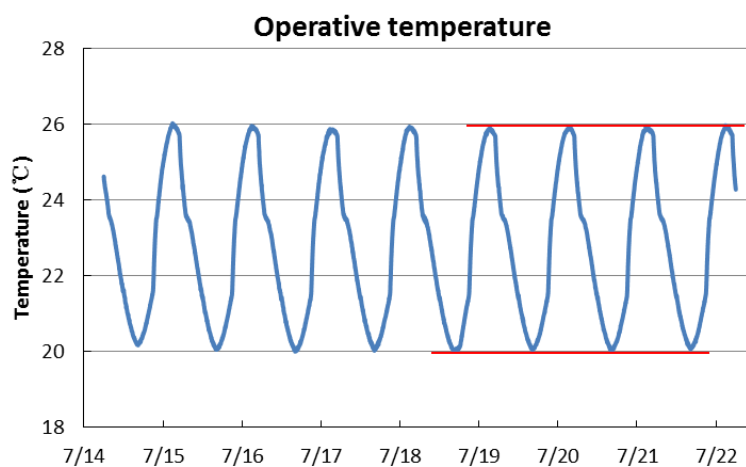


Figure 21 Cold box conditions.



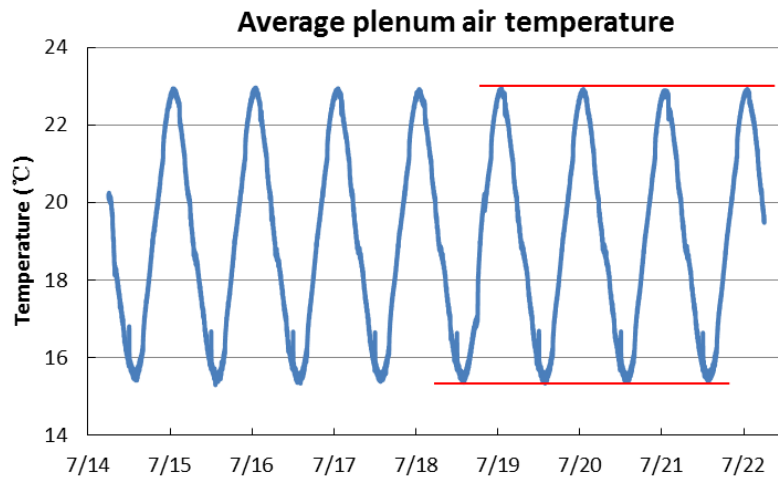
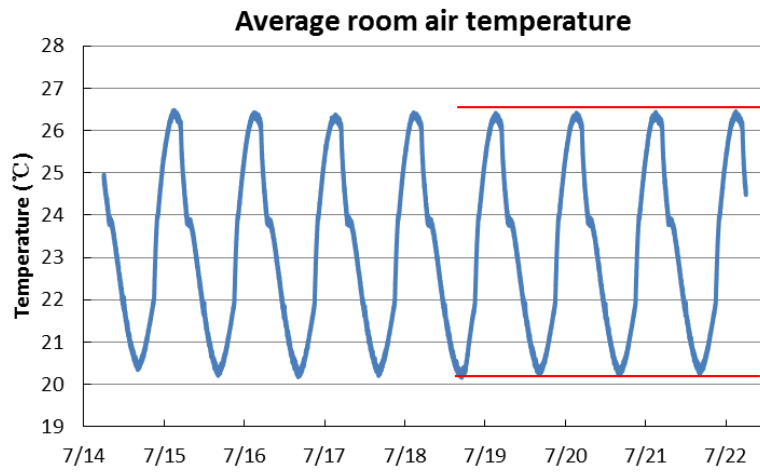


Figure 22 Room side conditions.

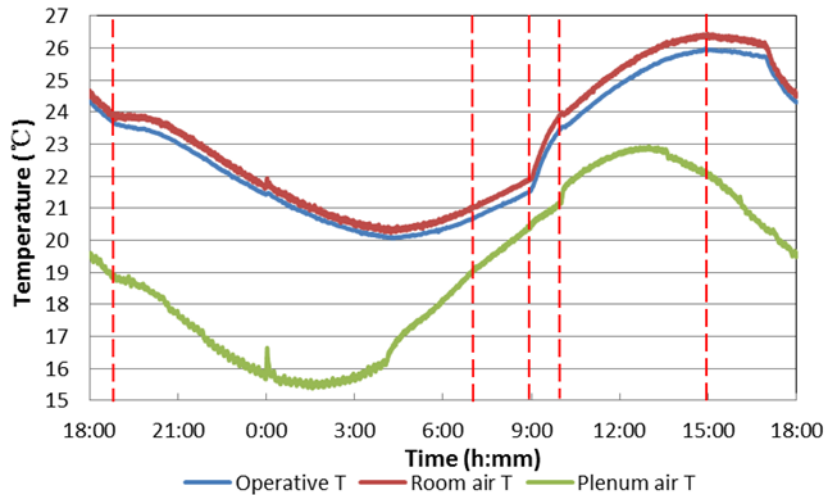


Figure 23 Quasi-steady state results of Case 3.

Figure 23 shows the results of the last day, it can be seen that the operative temperature is between 21.7 and 26.0 °C during the occupied hours, with the minimum at 9:00. The operative temperature increases very fast from the minimum to 23 °C, and during most of occupied hours it is within the comfort range in summer [4]. The ventilation system is activated during all day, with relative large ventilation rate.

Table 6 shows the control strategy used in this case. The measured ventilation rates are 216 m³/h and 369.7 m³/h for the low and high levels, corresponding to approximate 3 times and 5.1 times the minimum ventilation rate. The maximum ventilation rate used in this case is due to the consideration of the fan control. TABS are activated from 7:00 to 17:00, so totally 10 hours. The measured supply water temperature is 15.7 °C, and the constant water flow rate is 0.037 kg/s. Actually, the high ventilation rate in the plenum accelerates the response of TABS, it can be seen in Figure 24 that the return water temperature gets stable only after 3 or 4 days when TABS are activated.

Table 6 Control strategy used in Case 3.

Time	Ventilation	TABS cooling	Internal heat sources	Solar radiation
7:00	On (3*Min)	On	Off	On
9:00	On (3*Min)	On	On	On
10:10	On (5.1*Min)	On	On	On
17:00	On (5.1*Min)	Off	Off	On
19:40	On (3*Min)	Off	Off	On

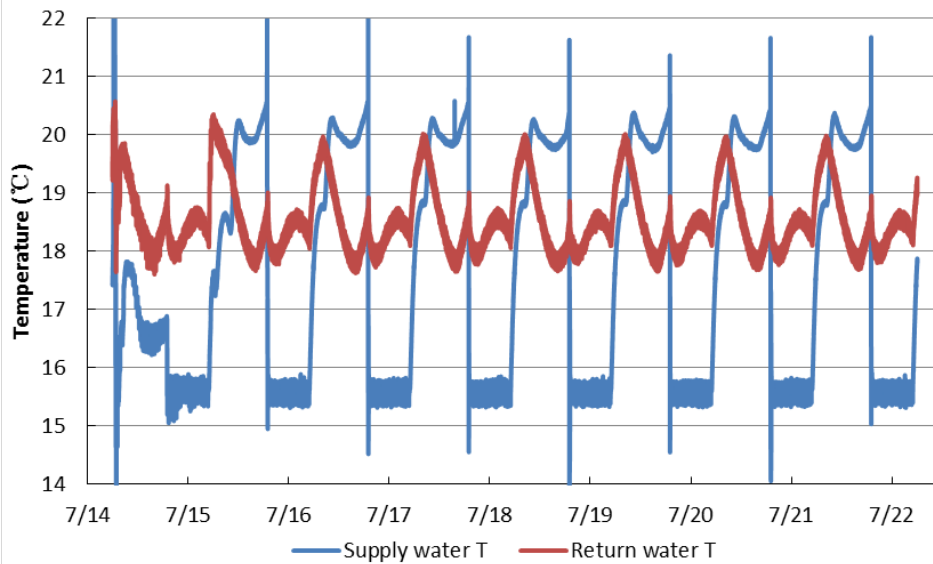


Figure 24 TABS water temperature in Case 3.

The measured power consumptions of heat sources are given in Figures 25 and 26, both with small changes with time. The average power consumption of all internal heat sources is about 480 W.

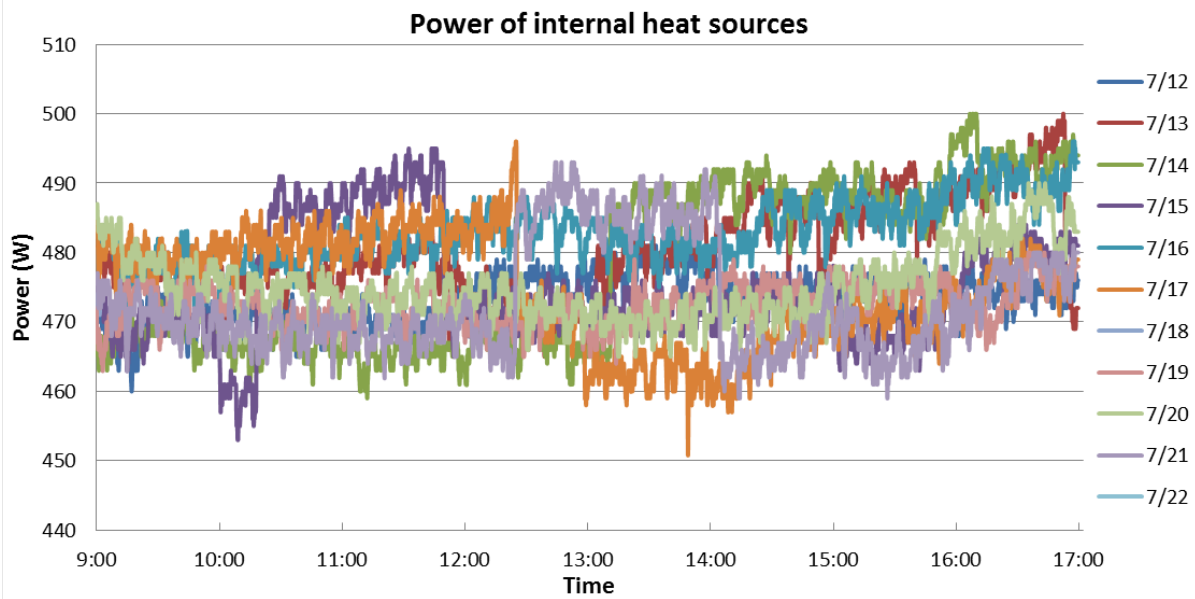


Figure 25 Power consumption of internal heat sources.

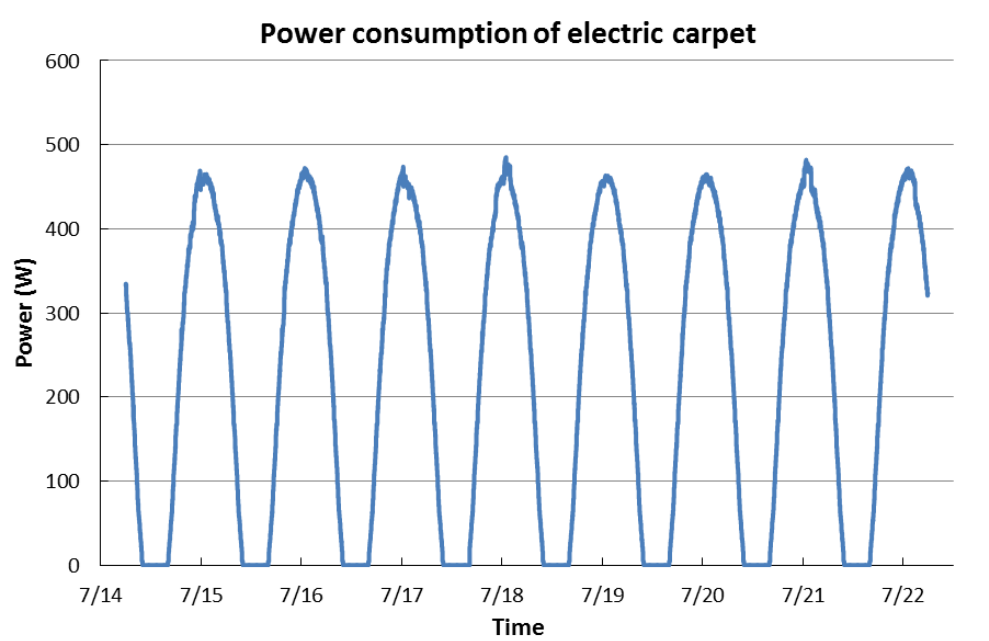


Figure 26 Power consumption of electric carpet.

7. Energy balance analysis

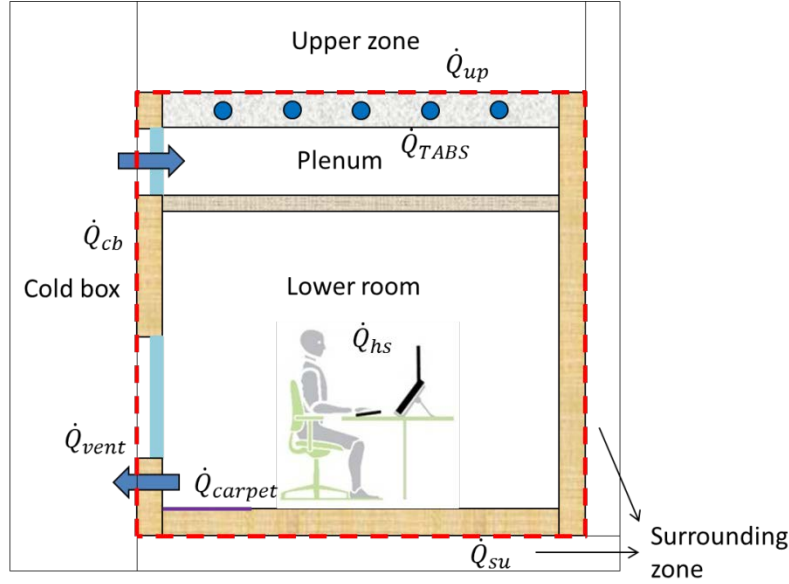


Figure 27 Thermal processes of the room.

Figure 27 shows the primary thermal processes in the room, including the heat gains from internal heat sources and solar radiation, heat loss/gain from TABS and ventilation, and heat transmission from the enclosed constructions. The space in the red rectangular area shows the room building thermal mass, including the facade, the floor, the interior walls, the diffuse ceiling, the concrete slab, the equipment and the air. The capacitance of air is very small, but it is still considered as part of the thermal mass in the energy balance analysis.

For the dynamic thermal processes, it is very hard to evaluate the energy stored and released time by time. Therefore, a total heat balance during 24 hours is considered for the evaluation of energy balance. The energy balance analysis has been carried out for a time step of 10 seconds and the total heat balance during 24 hours in a day is expressed by Equations. (2). The heat unbalance rate is defined in Equation (3). Assuming that the heat injected to the room is positive, and the heat removed from the room is negative.

$$\Delta\dot{Q} = \dot{Q}_{hs} + \dot{Q}_{carpet} + \dot{Q}_{vent} + \dot{Q}_{TABS} + \dot{Q}_{environment} \quad (2)$$

$$\bar{Q} = \frac{\Delta\dot{Q}}{\dot{Q}_{hs} + \dot{Q}_{carpet}} \times 100\% \quad (3)$$

$$\dot{Q}_{hs} = \dot{Q}_{manikins} + \dot{Q}_{lights} + \dot{Q}_{equipment} \quad (4)$$

$$\dot{Q}_{environment} = \dot{Q}_{up} + \dot{Q}_{su} + \dot{Q}_{cb} \quad (5)$$

Where,

$\Delta\dot{Q}$ is the error of the total heat balance, Wh/day.

\bar{Q} is the unbalance rate to the heat sources.

\dot{Q}_{hs} is the energy consumption of internal heat sources (two manikins, two desk lights, two computers, and two monitors), Wh/day.

\dot{Q}_{carpet} is the energy consumption of the electric carpet simulating the solar radiation, Wh/day.

\dot{Q}_{vent} is the ventilation heat transmission, Wh/day.

\dot{Q}_{TABS} is the energy delivered by TABS water to the system, Wh/day.

$\dot{Q}_{environment}$ is the heat exchange between the test room and the upper zone, the surrounding zone and the cold box, Wh/day.

Since the tests are carried out under quasi-steady state conditions, for a 24-hours energy balance analysis all terms in Equation (1) can be evaluated using the steady-state method as in Ref. [2].

Table 7 gives the results of heat balance analysis for all cases. The heat unbalance rates are within 10% except Case 2. The error can be attributed to the data logger resolution, the measurements of ventilation rate and water flow rate, the thermal properties of the facade with windows, the uncertain thermal bridge of the room enclosure, and so on. In Table 7 it also shows that there is always heat gain from the upper zone and heat loss to the cold box. Since the surrounding zone has a temperature very close to the room temperature, the heat gain from the surrounding zone is relatively small. In Case 1, the minimum ventilation brings the heat loss almost the same amount of all heat gains. Therefore, extra TABS heating should be supplied to the system, in order to keep the room temperature within the acceptable range. In Case 2, the total heat loss from ventilation can offset all the heat gains, so the room can get the neutral thermal conditions and no mechanical cooling is needed. In Case 3, the natural ventilation cooling potential is very limited even the ventilation is activated all day. TABS have to be activated to offset the remained heat gains.

Table 7 Energy balance of the room during 24 hours.

	\dot{Q}_{hs} (Wh/day)	\dot{Q}_{carpet} (Wh/day)	\dot{Q}_{vent} (Wh/day)	\dot{Q}_{TABS} (Wh/day)	\dot{Q}_{up} (Wh/day)	\dot{Q}_{su} (Wh/day)	\dot{Q}_{cb} (Wh/day)	$\Delta\dot{Q}$ (Wh/day)	\bar{Q} (%)
Case 1-no TABS	3800	0	-3737.5	0	1300.2	694.7	-2153.8	-96.4	2.5
Case 1- with TABS	3800	0	-3972.3	2335.4	272.4	242.5	-2311.5	366.5	9.6
Case 2	3800	3773.5	-8701.8	0	1303.0	158.7	-1313.7	986.3	13.0
Case 3	3800	5447.1	-7297.4	-4133.8	1813.6	261.9	-394.5	503.1	5.4

8. Thermal comfort analysis

According to ISO 7730:2005 [4], the thermal comfort in the occupied zone involves both whole-body thermal comfort and local thermal comfort. The indices of PMV and PPD can be used to evaluate the thermal comfort of the whole-body, and the operative temperature range is also available to evaluate it in the dynamic conditions. The local thermal comfort is assessed by draught risk, vertical temperature difference, radiant asymmetry, and so on.

8.1 Operative temperature range

Considering the dynamic measurements, initially, the thermal comfort analysis is based on the operative temperature during the occupied time and is evaluated according to standard EN 15251 [5]. Category II in EN 15251 is selected as it is in accord with the thermal comfort level ($-0.5 < PMV < +0.5$) in EN ISO 7730 [4]. A performance index (PI) associated with the category represents the percentage of values of operative temperatures during the occupied time that fall within the acceptable range of the category [6]. When the PI is at least 90%, the indoor thermal environment is supposed to meet a certain category.

In category II in EN 15251, the operative temperature of 20 °C - 24 °C is the thermal comfort condition in winter (Case 1), while the operative temperature of 23 °C - 26 °C is the thermal comfort condition in summer (Case 3). The adaptive thermal comfort according to standard EN 15251 [5] is used to evaluate the thermal comfort in transitional seasons (Case 2), the calculated adaptive comfort temperature range in Category II is from 19.4 °C to 25.4 °C.

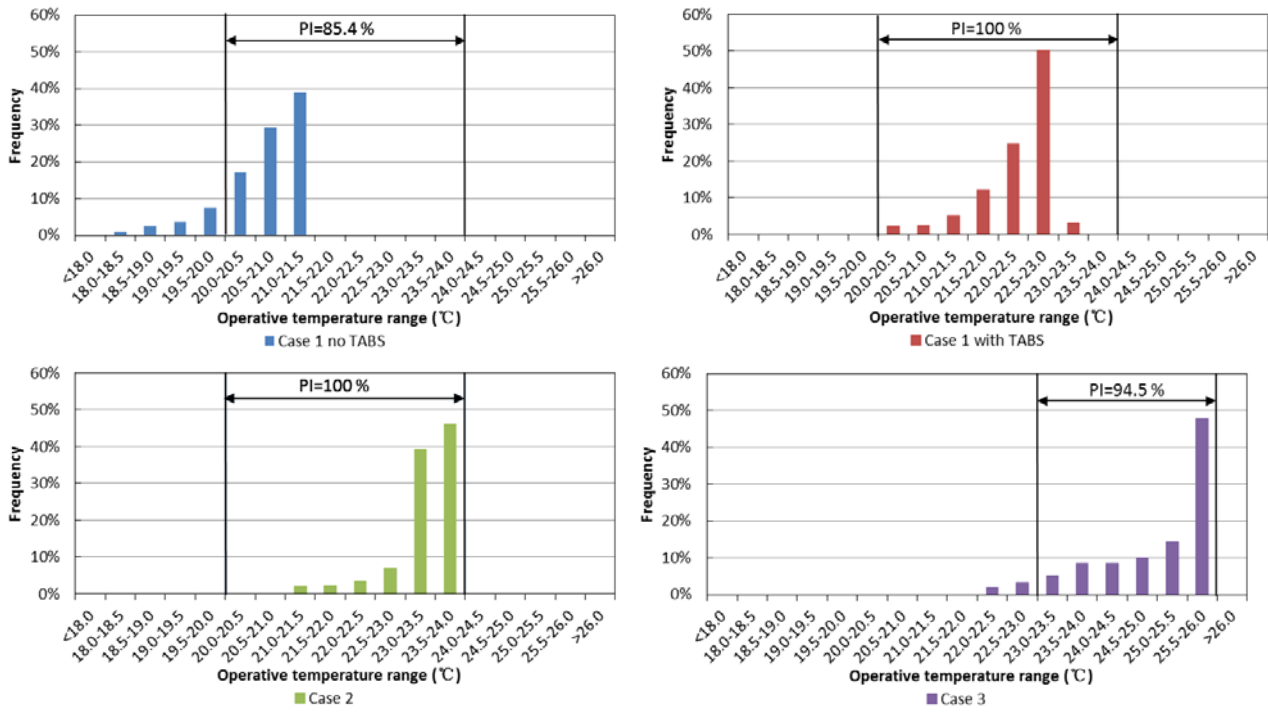


Figure 28 Thermal comfort of PI for the tests.

Figure 28 shows the results of thermal comfort based on the operative temperature range, the results are evaluated every 10 seconds. For the Case 1 without TABS heating, the PI is 85.4% so the thermal comfort is not acceptable. When TABS heating is added, the PI reaches 100% so the operative temperature is within the comfortable range. Case 2 and Case 3 also have the PI higher than 90%, and meet the comfortable range in the standards.

8.2 PMV/PPD

PMV and PPD are used to evaluate the whole body thermal comfort. The PMV is an index that predicts the mean value of the votes of a large group of persons on the 7-point thermal sensation scale, based on the heat balance of the human body. PMV is related to the thermal environmental variables, such as air temperature, relative humidity, mean radiant temperature and relative air velocity. Besides, it is largely dependent on the activity level and insulation of clothing of occupants. The PPD is an index that establishes a quantitative prediction of the percentage of thermally dissatisfied people who feel too cool or too warm [4], which can be calculated based on PMV.

In this study, the mean values of air temperatures and velocity magnitudes at the heights of 0.1 m, 0.6 m and 1.1 m can represent the thermal state of the whole body, and the PMV and PPD indexes are calculated based on these mean values. A metabolic rate of 1.2 and a clothing value of 0.5 are assumed in this calculation of PMV. The relative humidity in the room is measured by the two sensors at C-3 and C-5 (in Figure 5), at the height of 1.1 m. The mean radiant temperature is calculated from the plane radiant temperature according to ISO standard 7726 [7], which is determined based on the measured surface temperatures.

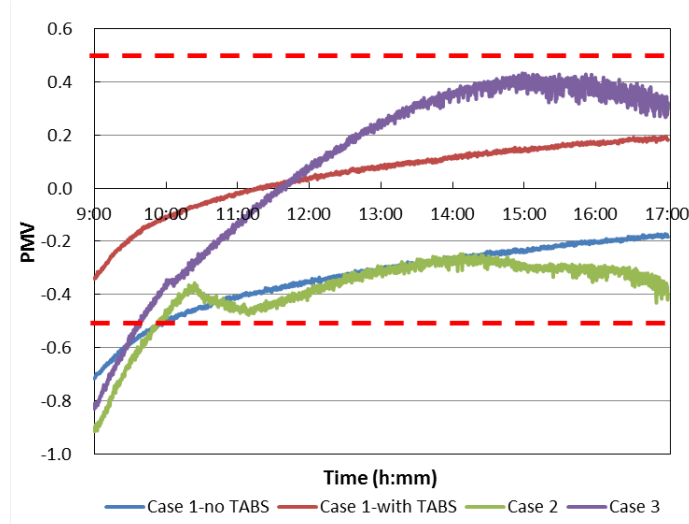


Figure 29 PMV results of all cases.

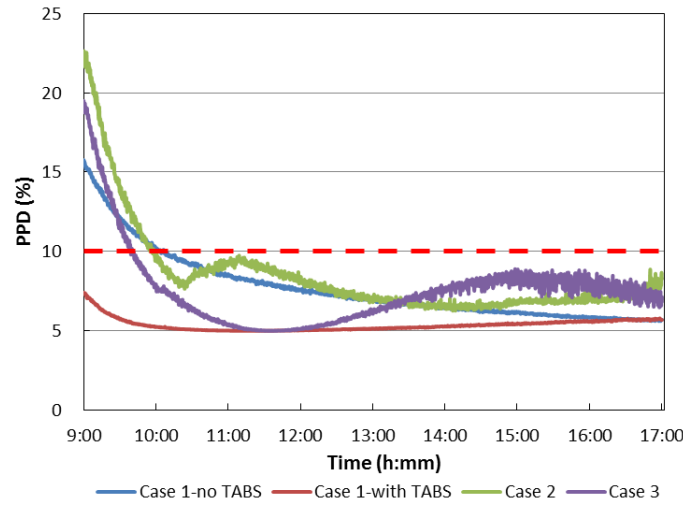


Figure 30 PPD results of all cases.

PMV and PPD results of all cases during the occupied hours are depicted in Figures 29 and 30. It can be seen that Case 1 with TABS heating has the best PMV and PPD. Since the surrounding surface temperature is relatively lower during the morning, the environment is a little cold for the other three cases with a PMV lower than -0.5 and PPD higher than 10% in the first hour. The time proportion of Case 1 without TABS heating when PMV is lower than -0.5 is about 13.8%, and this value for Case 2 and Case 3 is 11.6% and 8.1%, respectively. These results are very close to the evaluation in Section 8.1.

8.3 Draught rate (DR)

Thermal dissatisfaction can also be caused by unwanted cooling or heating of one particular part of the body, known as local discomfort [4]. The most common cause of local discomfort is draught, which can be expressed as the percentage of people predicted to be bothered by draught and can be calculated by Equation (6).

$$DR = (34 - t_{a,l})(\bar{v}_{a,l} - 0.05)^{0.62}(0.37 \cdot \bar{v}_{a,l} \cdot T_u + 3.14) \quad (6)$$

For $\bar{v}_{a,l} < 0.05$ m/s, use $\bar{v}_{a,l} = 0.05$ m/s.

The draught rate (DR) in this study is calculated at a height of 0.1 m, 1.1 m and 1.7 m at the locations of C-1, C-2, C-3, C-4, C-5 and C-6 (in Figure 5), respectively. A constant turbulence intensity of 40% is assumed in the DR calculation.

Table 8 Maximum DR during the occupied hours (unit: %).

	Case 1- no TABS			Case 1- with TABS			Case 2			Case 3		
	0.1 m	1.1 m	1.7 m	0.1 m	1.1 m	1.7 m	0.1 m	1.1 m	1.7 m	0.1 m	1.1 m	1.7 m
C-1	0	0	0	0	0	1.2	6.0	8.2	7.7	6.0	3.8	5.7
C-2	5.2	2.2	0	4.5	3.6	0	6.8	5.3	7.2	7.0	3.3	6.1
C-3	5.7	2.5	0	4.6	2.2	0	5.7	6.1	8.0	5.3	4.8	3.9
C-4	2.2	0	0	0	0	0.9	4.8	4.4	6.9	5.1	3.1	4.8
C-5	0	1.3	12.4	0	0	9.8	9.0	3.6	11.3	7.1	2.4	10.6
C-6	0	0	0	0	0	0	9.7	0	1.5	7.7	0	0

The DR results in Table 8 show that all positions have very good thermal comfort with almost all maximum values of DR below 10%, corresponding to the Category A in EN 7730 [4]. The largest DR is found at the height of 1.7 m at C-5, which may result from the heat sources. Actually, the internal heat sources are close to this position, so the thermal plume may be very strong at this height. Comparing Case 1 and the other two cases, when there is solar radiation the local discomfort increases with higher values of DR.

8.4 Vertical air temperature difference between head and ankles

A high vertical air temperature difference between head and ankles can cause discomfort [4], which can be evaluated in Equation (7).

$$PD = \frac{100}{1 + \exp(5.76 - 0.856 \cdot \Delta t_{a,v})} \quad (7)$$

$\Delta t_{a,v}$ is vertical air temperature difference between head and feet, °C.

Table 9 Maximum PD of vertical air temperature difference during occupied hours (unit: %).

	C-1	C-2	C-3	C-4	C-5	C-6
Case 1- no TABS	0.8	0.8	1.2	1.0	1.1	0.2
Case 1- with TABS	0.9	0.9	1.1	0.9	1.3	0.2
Case 2	0.4	0.7	0.8	0.9	1.7	0.3
Case 3	0.6	1.1	1.2	1.5	2.1	0.3

The maximum PD during occupied hours due to the vertical air temperature difference between the height of 0.1 m and 1.7 m is given in Table 9. The PD caused by the vertical air temperature difference meets the Category A in EN 7730 [4]. The maximum is found at C-5 in all cases, which may result from the heat sources as the reason described in Section 8.3.

8.5 Warm or cold floor

If the floor is too warm or too cool, the occupants could feel uncomfortable owing to thermal sensation of their feet. For people wearing light indoor shoes, it is the temperature of the floor rather than the material of the floor covering which is important for comfort [4]. The PD caused by the warm or cold floor can be calculated using Equation (8).

$$PD = 100 - 94 \cdot \exp(-1.387 + 0.118 \cdot t_f - 0.0025 \cdot t_f^2) \quad (8)$$

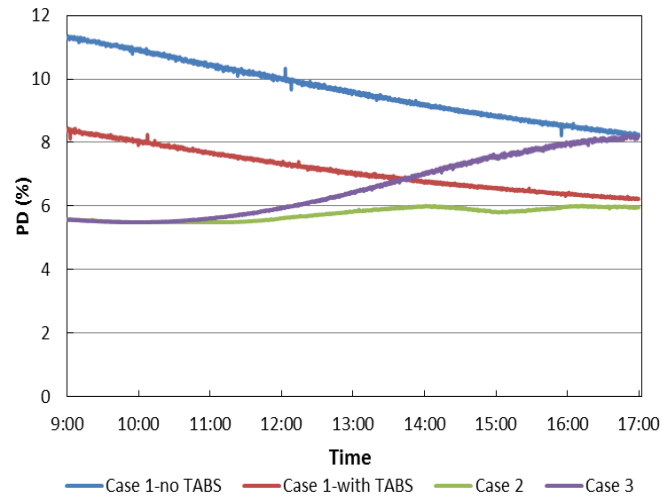


Figure 31 PD caused by the warm or cold floor.

Figure 31 depicted the PD caused by the warm or cold floor. Except the Case 1 without TABS, PD values of all cases during the occupied hours are within 10%.

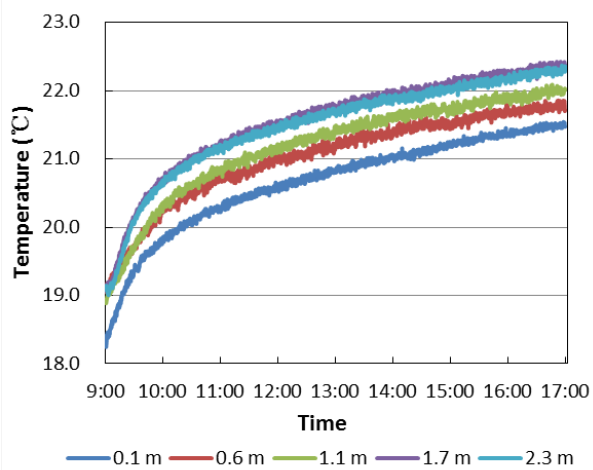
8.6 Radiant asymmetry

Radiant asymmetry, resulting from the different surrounding surface temperatures, can also cause discomfort. People are most sensitive to radiant asymmetry caused by warm ceilings or cool walls (windows) [4]. The radiant temperature asymmetry is estimated as the difference between the plane radiant temperatures in two opposite directions. As described by Fanger [8], it refers to a small plane element 0.6 m above the floor (the height of the center of a seated person) and horizontal to characterize radiant asymmetry caused by a warm or cool ceiling. Due to the effect of diffuse ceiling, the direct radiant effect of TABS slab surface on the room space is reduced. So the ceiling lower surface temperature is close to the other surfaces, and the radiant asymmetry is decreased. As stated in ISO 7730, occupants will feel discomfort when the radiant asymmetry is larger than 5 °C by warm ceiling, and larger than 14 °C by cooling ceiling. In our dynamic measurements the ventilation first cools the plenum, so the surface temperature of the diffuse ceiling panel is relatively lower than that of the lower room surfaces. Thus, the diffuse ceiling can always be considered as a cold ceiling. The maximum asymmetry in the dynamic measurements is 1.4 °C found in Case 2, corresponding to the maximum PD caused by the radiant asymmetry is within 0.01%. Therefore, there is no influence of radiant asymmetry on the local thermal comfort in the dynamic measurements.

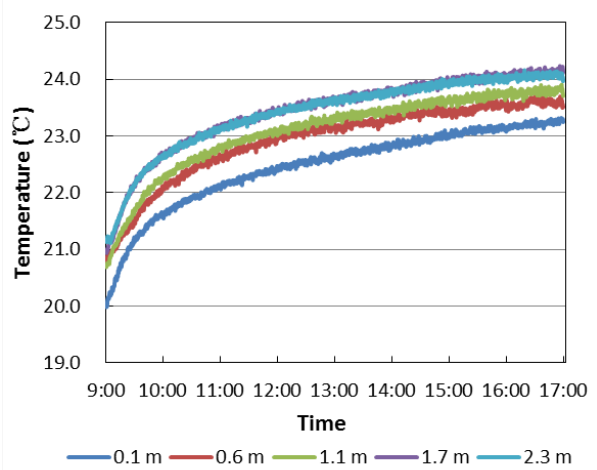
8.7 Vertical air temperature and air velocity distributions

Air temperatures and air velocities at six positions are measured with anemometers at the heights of 0.1 m, 0.6 m, 1.1 m, 1.7 m and 2.3 m, respectively. The average results of six positions at every height are presented in Figures 32 and 33.

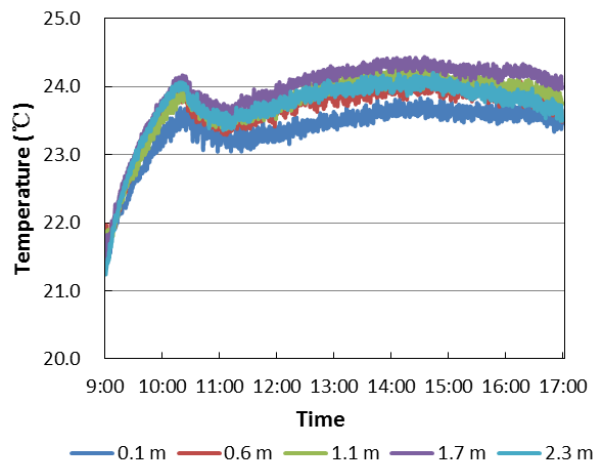
Figure 32 shows that there is a maximum of 1 K temperature difference from the floor to the ceiling in all cases, corresponding to a temperature gradient of less than 0.5 K/m. This indicates a good mixing of the room air when using diffuse ceiling ventilation, which shows good agreement with the findings by Nielsen et al. [9] and Fan et al. [10]. The temperature at the height of 2.3 m is a little lower than that at the height of 1.7 m, which may be the effect of diffuse ceiling. In order to bring sufficient cooling, the air in the plenum is always colder than the room air. This cold air will first penetrate through the diffuse ceiling panel and then mix with the air at the upper part of the room, so the air at the height of 2.3 m is cooled down.



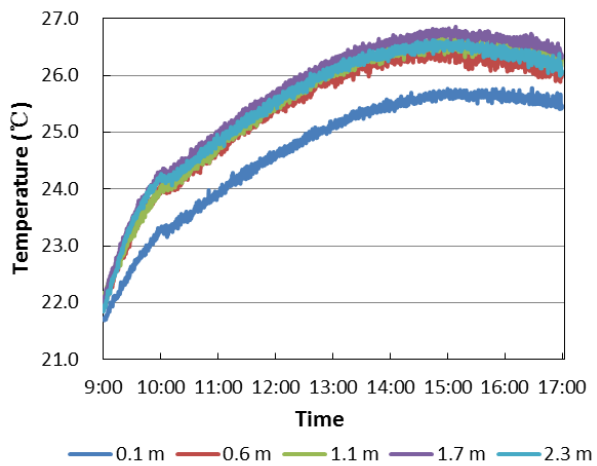
Case 1-no TABS



Case 1- with TABS

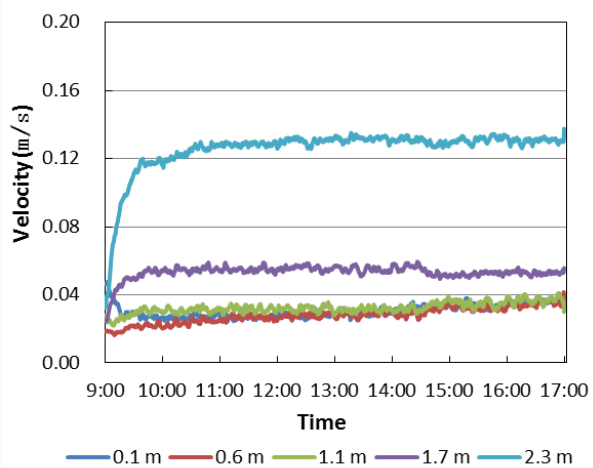


Case 2

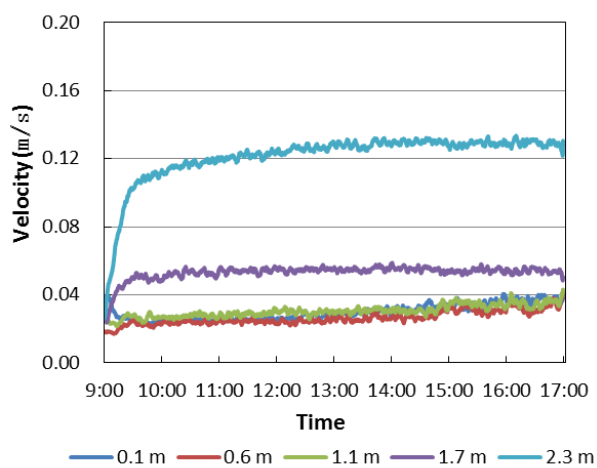


Case 3

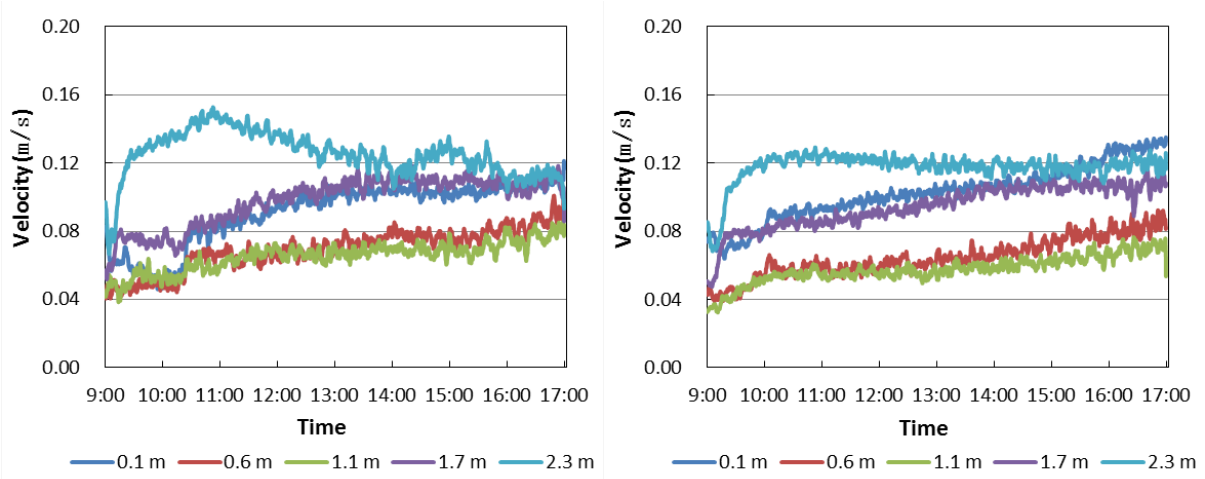
Figure 32 Vertical air temperature distributions.



Case 1-no TABS



Case 1- with TABS



Case 2

Case 3

Figure 33 Vertical air velocity distributions.

As shown in Figure 33, the measured velocity magnitudes for all case are lower than 0.15 m/s. The difference in air speed from ankle to head level is within 0.05 m/s. In Case 1, the velocity in the occupied zone is very stable and very low. While in Case 2 and Case 3, due to the increase of ventilation rate, the air velocities in the occupied zone increases. Moreover, the air velocities at the height of 0.1 m and 1.7 m are very close. This may be caused by the heat sources, which generates the strong thermal plume at the height of 1.7 m. Due to the momentum of the high ventilation rate the downward flow drops to the floor later. Thus, the velocity at the height of 0.1 m is relatively high and close to the velocity at the height of 1.7 m.

9. Comparisons with building simulations

A building simulation model representing the whole hot and cold chambers is built in BSim, which is used to compare and validate the experimental results. The built model is shown in Figure 34, with the same dimensions of the experimental chamber. The measured air temperatures in the cold box, upper zone and surrounding zone are given as the boundaries in the building simulations, and the measured power consumption of all heat sources are used in the building simulations as well.

The compared results are depicted in Figures 35-38. ‘M’ means the results measured and ‘S’ indicates the results simulated. The results show that the building simulation tool-BSim cannot well simulate the building thermal processes in the measurements, with relatively large deviations of the temperatures. The deviations may result from two primary reasons. First, BSim cannot simulate the ventilation through the diffuse ceiling, which could cool down the diffuse ceiling first. Instead, the diffuse ceiling is simulated as an ordinary ceiling panel with the measured thermal properties [2]. The ventilation through the ceiling panel is just simulated as the mixing ventilation for the lower room space, and the air in the plenum is used as the source of this mixing ventilation. Since this thermal process is very important, it influences the energy balance of the plenum and the lower room space. On the other hand, the internal heat sources are designed from 9-17, but the solar radiation simulated is activated dynamically. BSim is based on the hourly simulation, thus it is hard to determine the same amount of solar heat gain at the same time of measurements. Based on the above two points, due to the absent modelling of diffuse ceiling ventilation and the hourly simulation in BSim, it cannot predict the energy performance of the building with the proposed system accurately. Therefore, it is better to investigate an accurate model for the diffuse ventilation with a smaller time step in the future building simulations for this kind of buildings.

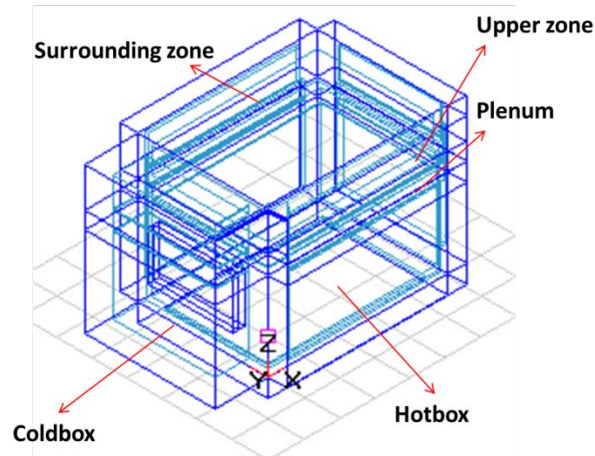


Figure 34 BSim model of the experimental hotbox.

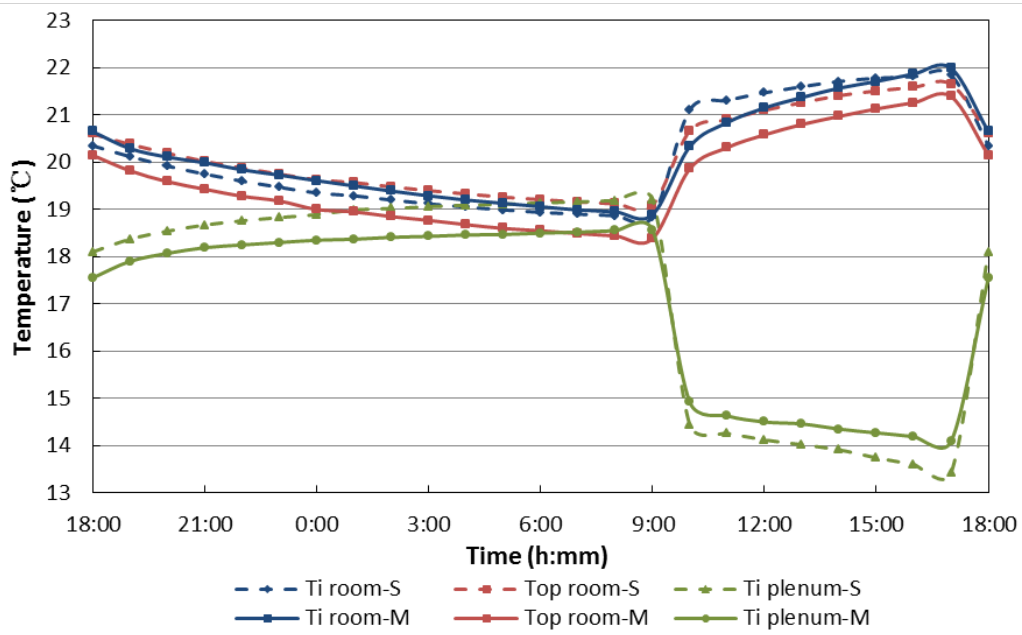


Figure 35 Comparison of Case 1 without TABS heating.

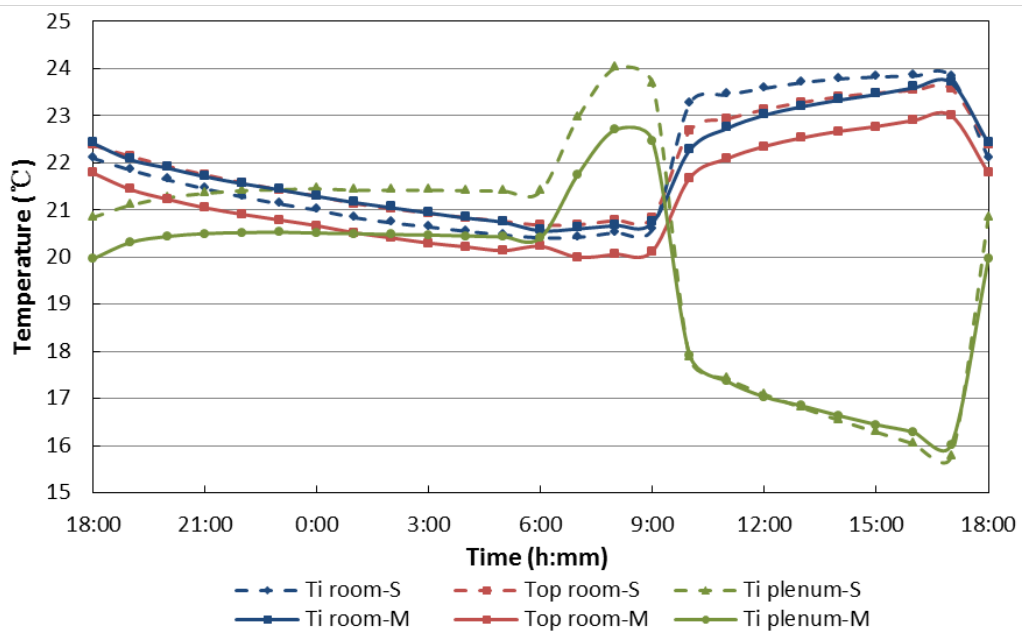
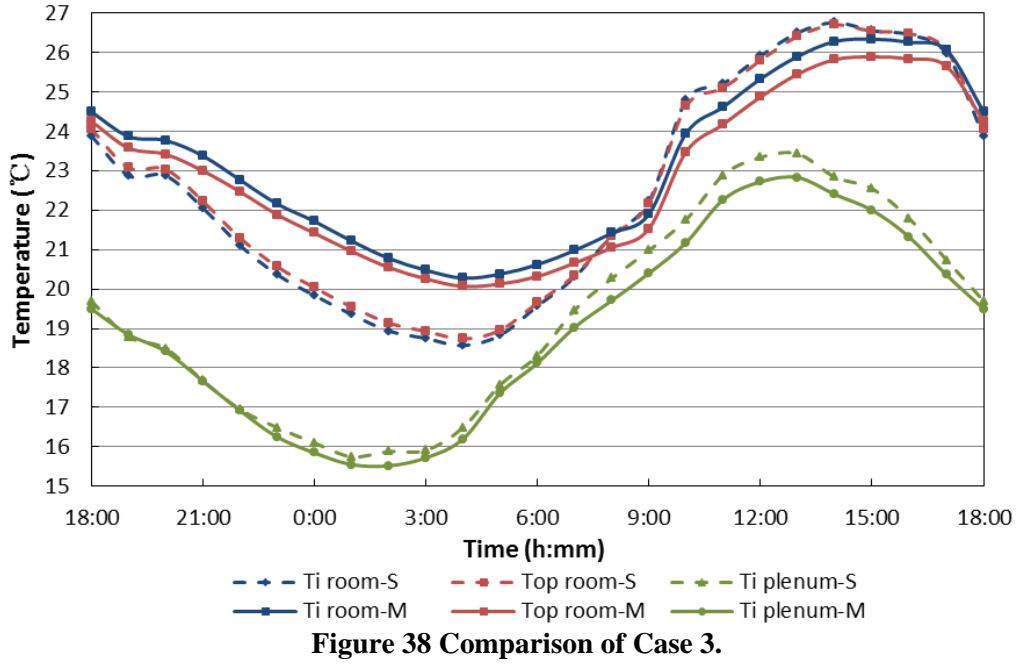
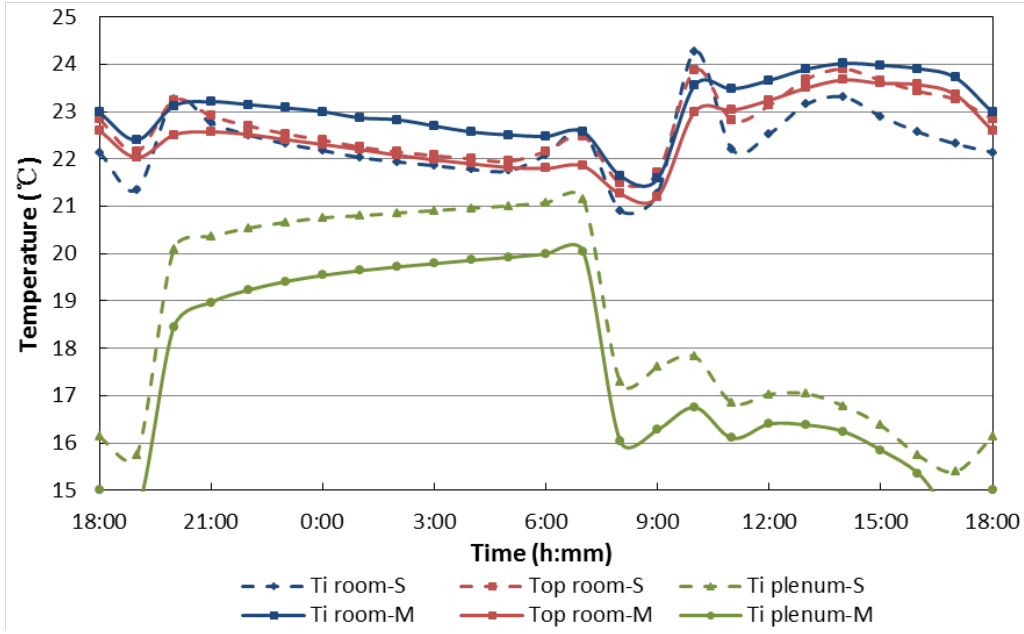


Figure 36 Comparison of Case 1 with TABS heating.



10. Discussions and Conclusions

This report presents the dynamic measurements of a novel HVAC system combining natural cooling with diffuse ceiling inlet and TABS, which are carried out in the Hotbox in Aalborg University. Three typical weather conditions are selected, including typical winter, transitional seasons and typical summer. Both outside air temperature and solar radiation in each case are considered changing during day and night. The test room is set up to simulate a real office environment. Different control strategies are tested, the main purpose is to use the natural ventilation cooling to the maximum level. All cases are carried out under the quasi-steady state, and it takes several days to get the quasi-steady state. Finally, the energy balance and the thermal comfort of all cases are analyzed.

The test results show that the present control strategy used in each case can ensure a good thermal environment in the test room. The energy balance indicates that the energy storage of the room

thermal mass makes a good use of ventilation cooling. In the winter case, extra TABS heating is needed even with the minimum ventilation rate. In the transitional case, natural ventilation is very suitable to keep a perfect thermal environment in the room without any mechanical cooling. However, in the summer case, the natural cooling capacity is insufficient even the night ventilation is used. Therefore, extra TABS cooling is indispensable.

The thermal comfort analysis shows that the entire thermal comfort is quite good with an acceptable operative temperature range during the occupied hours. The local thermal comfort is also quite good, without any high PD resulting from the influencing factors. Due to the application of diffuse ceiling, the vertical temperature gradient is very low. Meanwhile, the low vertical air velocity is also beneficial to keep a comfortable environment in the room.

The measured results are compared with the BSim simulations, and the results show some large deviations of the temperatures. The primary reasons for these deviations may be the absent modeling of diffuse ceiling ventilation and the hourly-based simulation in BSim. Therefore, in future it is better to study the modeling of the diffuse ventilation in the building simulation tools, so that the building simulation tools can be used to predict the energy performance of this combined system accurately. Besides, the optimum control strategy can also be studied through the improved modeling of the building simulation tools.

Through testing the control strategies used in different conditions, we know more knowledge about the dynamic thermal processes of the room with the proposed systems. Although the thermal comfort in different climatic conditions can be ensured when using the proposed control strategies, some optimization work still needs to be further studied. Both natural ventilation and TABS can supply the cooling, the optimum combination of them can lead to the minimum energy use. In the tests, actually two extreme weather conditions are considered, the winter without any solar radiation and the summer with a huge amount of solar radiation. A good suggestion is that when using this system in winter it is better to use the solar radiation as much as possible, while in the summer it is better to limit the solar radiation to the room. When TABS have to be used, it is better to determine when and how to activate it, just to find the optimum control strategy with the minimum energy use and an acceptable thermal environment.

Since cooling potential of natural ventilation is of great importance in this system, this system is highly dependent on the climatic conditions and the room building thermal mass. A good suggestion is to use this system in a temperate climatic condition with high natural ventilation cooling potential, where the activation of TABS can be reduced. When this system is to be used in a building, it is better to perform the energy simulation to investigate the natural cooling potential at the design stage.

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